POWER

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IN the New York daily press McGill's cartoons of the trials of the "Economic Husband" in his efforts to practise economy afford much amusement. There is a field just as fertile for the humorist to be found in the "economies" sometimes proposed when it is decided to install the heating system for the new plant.

A certain superintendent of motive power and the chief engineer of a large system were discussing the merits of hot-water versus vacuum heating for a large shop plant with the salesmen.

"If there were zero weather all of the time," said the hot-water man, "there would be but little difference in the economy of the respective systems. With the average weather conditions, about 30 degrees, I could show better economy with hot water."

When the chief engineer asked the vacuum-heating salesman what he could show for his system in moderate weather, he replied:

"We have never made any observations for *average weather;* we only consider the *coldest weather.*"

It is intended in the hot-water heating article in this issue to emphasize the point that although extreme weather conditions should be given consideration, the average weather, which occurs the greater portion of the year, is the time when a possible saving in operation can be effected.

In the majority of cases, the only condition taken into account when the design of a plant is under consideration is the amount of steam necessary for heating in extreme weather, and the exhaust steam for this purpose is regarded entirely as a byproduct.

This article shows that the power should be the byproduct, as the steam for heating is fixed by the requirements whether the engines are operated or not, and 90 per cent. of the heat of steam at any pressure is required to vaporize the water.

Aside from the steam employed for manufacturing processes, in most every instance, if the steam for heating under zero-weather conditions were used to best advantage in an engine, considerably more power for the plant would be available.

Where steam is used for heating the impression is general that it makes little difference whether 5 or 10 horsepower are obtained when passing it through the engine.

In 99 cases out of 100, all of the machines are connected to a general exhaust pipe, with one end to the heating system and the other with a relief valve to the atmosphere. In moderate weather back pressure is created on all units up to the limit at which the relief valve is set. If part of the units were exhausted to the atmosphere and only those used upon the heating system determined by the requirements of the weather, back pressure on the whole system would be relieved in many cases.

The article was written by a practical man thoroughly familiar with the subject; although introducing the comparatively new subject of hot-water heating by forced circulation, a brief study of the curves and tables will make plain many of the complications arising in the use of exhaust steam in heating plants in general.

Developments in Prime Movers

By Prof. A. G. Christie

European practice in boilers, steam piping and auxiliaries as viewed by an American engineer and compared with our present practice. A future instalment will deal with turbines and generators.

Although it is interesting and profitable to study the development of prime movers and auxiliaries of European make, one cannot predict that similarly designed machines would meet with equal success in America on account of the different economic conditions and also because of the dissimilar manufacturing methods.

Boilers and Furnaces

Outside of Great Britain, the water-tube boiler seems to have been universally adopted in Europe. Attention is now being paid to the boiler settings, and the marine type of boiler with its closed sheet-steel casing is rapidly gaining favor. Fig. 1 shows a boiler of this type having 4780 square feet of heating surface with 1580 square feet of superheating surface and 3082 square feet of economizer surface. A chief engineer in one of the leading English power stations informed the writer that they figured on 5 per cent. increase in the economy of boilers when steel casings are used, due to the elimination of air leakage. The boiler shown in the illustration is installed in one of the leading central stations of Berlin, and under test gave efficiencies over 80 per cent., including the economizers. Without the economizers the efficiency ranged from 75 to 79 per cent. at various loads.

Attention is called to the steam drum above the boiler proper. This is quite common practice with German boilers. In Germany the practice of welding certain parts of the boiler has become established; for instance, the Borsig Company welds the water legs of its boilers to the main drums instead of riveting them, as is the practice in the United States. The sheets of the Borsig boilers are butted along the axial seams and are riveted with butt straps. The sheets themselves are welded together for about 1 foot from each end, so that there is no thinning out of the plate for the lap where more than one end of the sheet comes together. With this type of boiler the path of the gases before reaching the heating surface is very long, and with the firebrick arch provided almost smokeless combustion is obtained under all conditions of load. Another decided advantage of this setting is that it is possible to have doors all around the furnace thereby giving access to any part for the purpose of breaking up clinkers. These doors are provided with smoked-glass peepholes so that the firemen can note at all times the condition of the fire without opening the door.

In Germany an auxiliary grate at the rear of the main chain grate is used (see Fig. 1). When the load is suddenly increased, requiring heavier feeding of coal at the front of the stoker, the speed of the stoker is increased. Under these conditions frequently large quantities of unburned coal are carried over into the ashpits from the rear end of the grates. With the auxiliary grate, however, this coal is held until it is completely burned and is then dropped into the ashpit. Two pits are also provided with most of these furnaces, one for the small screenings of coal which fall through the grates and the other for the ashes and refuse. Most of the stokers have been so developed that they handle the low-grade coals quite as readily as those of higher grades.

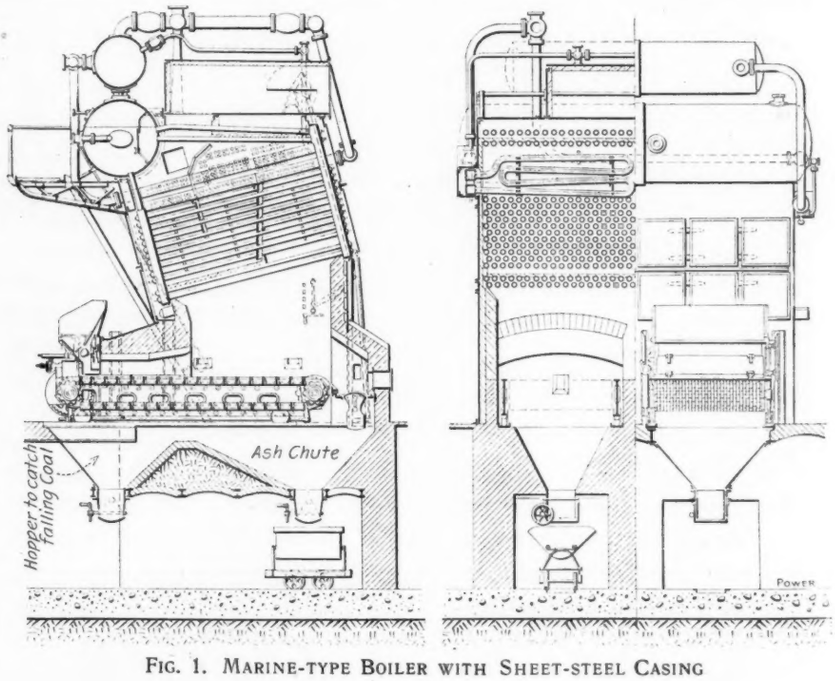


Fig. 1. Marine-type Boiler with Sheet-steel Casing

In Europe, as in many parts of the western States, there occur large deposits of coal which contain considerable ash and slate. This would furnish a very cheap supply of fuel in many cases provided it could be burned without trouble. Usually, however, the ash contents are such that they form heavy clinkers and so destroy the furnace that their use is prohibitive in ordinary operations. While many attempts have been made so far to burn this class of coal, but without much success, the utility of powdered coal in the rotary kilns of cement mills has encouraged inventors to adopt the same methods in burning coal high in ash. The difficulties met with in burning coal in such a manner may be summarized as follows: First, the difficulty of maintaining continuous and steady ignition. This is overcome only by maintaining a uniform and very high temperature in the furnace. Second, the failure to find an economical material to stand the destructive temperature necessary in the furnace. Third, the difficulty of maintaining a homogeneous mixture of the fuel dust and air for the full period required for combustion. The larger particles naturally fall more quickly to the bottom of the furnace; hence are in contact with the air for a shorter period of time and are not completely burned. This can only be overcome in part by extremely fine grinding, which would make the cost prohibitive. Fourth, there is difficulty in handling the molten ash which, due to the high temperature, usually forms a solid slag.

These obstacles have been overcome apparently by the Bettington boiler, designed to handle the low grades of South African coal and now being built by the Frazer & Chalmers Company, of London. This is shown in Fig. 2. Air is drawn by means of a fan through an

air heater placed in the path of the waste gases from the furnace and is there heated to a temperature ranging from 200 to 300 degrees Fahrenheit.

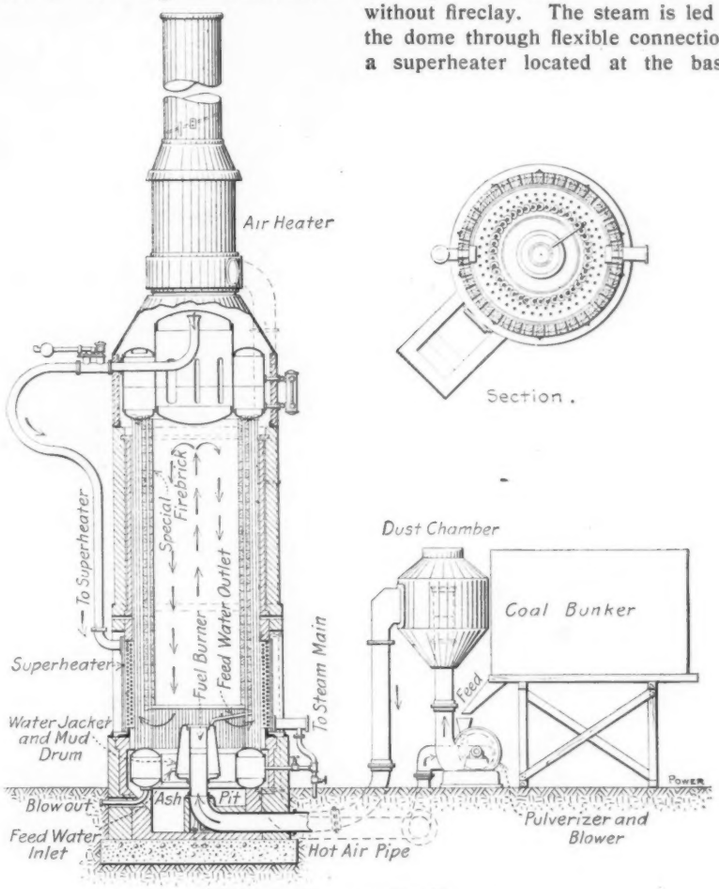


FIG. 2. BETTINGTON BOILER

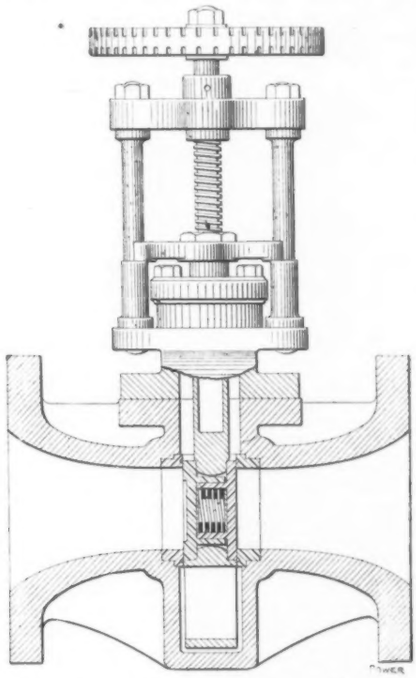


FIG. 3. HOPKINSON-FERRANTI VALVE

This fan is of peculiar construction, and serves both as a blower and a pulverizer, being fitted with manganese-steel blades which crush the coal. The air temperature is sufficient to evaporate most of the moisture in the coal and the fine coal dust mixed with air forms an explosive mixture. This passes to a dust chamber where the heavier particles separate out and are returned to the pulverizer. The finer dust particles are carried along with the air through the tuyere pipe to the burner. The mixture takes fire at this point, resembling closely the flame from the burners of a rotary kiln in a cement mill. The column of burning fuel passes up through the central part of the boiler and then is deflected downward around the sides after hitting either the cold-water drum above or a cushion of cooler gases next to the drum. It then passes up through the outer row of tubes around the upper drums and then through the air preheater to the stack.

The boiler itself consists of a water-jacketed blast pipe into which the feed water is fed. This water then passes to the upper drum as shown. This drum consists of mild-steel headers, with a central portion forming a steam chamber, and is connected with the lower ring header by the concentric rings of tubes. The inner set of tubes is lined with special C-shaped firebrick set up without fireclay. The steam is led from the dome through flexible connections to a superheater located at the base of the boiler, after which it passes to the steam main.

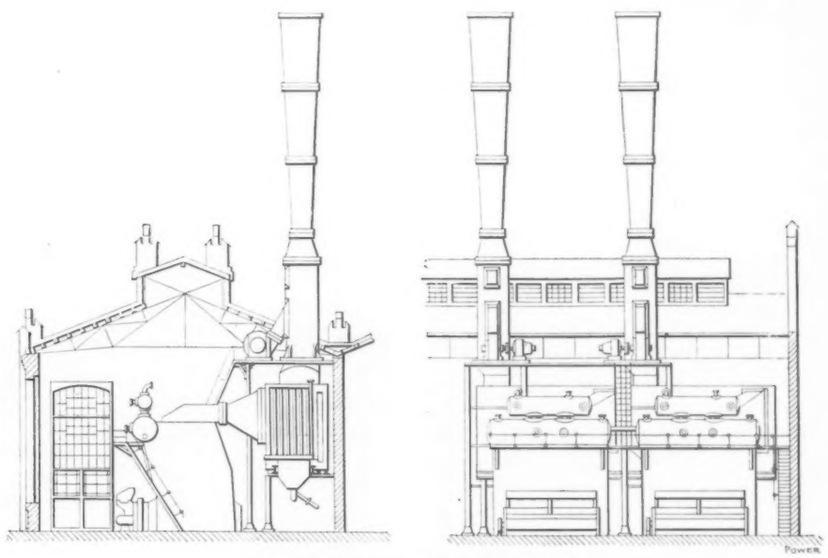


FIG. 4. EJECTOR FORM OF STACK

Combustion starts as the mixture leaves the burner, the coal-dust particles burning instantly, and the fine ash is carried off as dust. The heavier particles catch fire and continue to burn owing to the continually changing contact of each particle with a fresh supply of oxygen. On account of this continuous mixing of fuel and air during combustion very little more air than that theoretically necessary for complete combustion need be supplied. Hence very high temperatures occur at the middle of the furnace and the ash portion of the fuel is converted into a liquid spray which is projected against the firebrick walls of the boiler. These walls fuse into one solid mass from top to bottom and are only prevented from melting down entirely by the cooling action of the tubes which they inclose. When the molten ash builds up to such a thickness that the tubes cease to exert sufficient cooling effect, it then drops to the bot-

tom of the lining and into the ashpit in pieces about the size of a marble. This action seals all joints or cracks in the setting and prevents leakage. The intense heat to which the inner row of tubes is subjected circulates the water very rapidly and there is, therefore, very little scale deposited in these tubes, making them less liable to failure than the other tubes.

TABLE 1. RESULTS OF TESTS ON BETTINGTON BOILER

Coal used	Broomhill Bituminous Nutty	Broomhill Bituminous Slack	Dant: Clifton and Kersley	Dant: Ellesmere	South African: Partings	South African: Pickings	South African: Duff	Welsh: Anthracite Duff
Analysis (on dry coal)								
Ash	14.33		23.00	21.12	22.58	21.30	19.92	12.64
Volatile matter	32.55		20.00	26.46	27.38	27.68	22.94	11.66
Sulphur	0.99		.....	3.13	0.85	0.95	0.76	0.99
Fixed carbon	52.13		57.00	52.42	50.04	51.02	57.14	75.70
Calorific value on dry coal (B.t.u.)	11,870		10,700	11,250	10,098	10,494	11,088	12,276
Pounds of water evaporated from and at 212 degrees Fahrenheit	12.29		11.08	11.65	10.45	10.86	11.48	12.71
Absolute steam pressure, pounds per square inch	165		164.5	164.8	152.3	152.15	130.45	151.1
Average temperature gases escaping from boiler, degrees Fahrenheit	580	750	592.7	658.8	558.3	587.5	603.7	560.3
Average temperature feed water, degrees Fahrenheit	60	64	.....	56.6	56.0	56.0	56.0	57.2
Pounds of dry coal fired per hour	897	2,733	1,020	1,179.2	1,183.3	1,347.8	1,183.5	1,242.2
Actual water fed per hour, pounds	6,828.6	17,307	6,550	7,984	7,348.1	8,803.6	8,471.4	9,217
Equivalent water evaporated per hour from and at 212 degrees	8,714*	22,374†	8,476.8	10,290.4	9,437.2	11,321.4	10,909.5	11,776.4
Pounds of water per pound coal, actual	7.61	6.33	6.418	6.635	6.111	6.401	6.986	7.36
Pounds of water per pound of coal, from and at 252 degrees Fahrenheit	9.71	8.18	8.309	8.726	7.975	8.40	9.218	9.48
Thermal efficiency based on coal, per cent	79	66	75	74.9	76.31	77.35	80.30	74.59
Percentage of moisture in coal during test	3.85	3.85	15.6	2.00	1.6	2.00	2.4	0.8

*Normal rate. †High rate.
NOTE.—Temperatures range from 150 to 200 degrees less than the above when past the air heater. Heat returned to the furnace by air blast.

The amount of steam generated can be regulated by adjusting the supply of coal and air to the pulverizer. With the air preheater, it is said that coal with 15 per cent. or more moisture can be burned with ease. The pulverizing and grinding are said to require a power consumption equivalent to 3 or 4 per cent. of the boiler capacity.

Table 1 gives the results of some tests on this type of boiler. Attention is called to the high ash content of the coals used and also to the moisture in the third test.

STEAM PIPING

In general it may be said that the steam-piping systems in European power plants are of inferior design to those in our large central stations, but, on the other hand, they are usually kept in excellent repair. European practice differs in several particulars from that in America. For instance, no cast-iron or semi-steel fittings are used where the pressure exceeds 100 pounds, with the exception of some very old plants in the north of England. Above 100 pounds pressure, cast steel is invariably used. The bends used in these pipe lines were of very poor design, and leaky steam joints frequently caused trouble. In the main steam lines, globe valves are almost invariably used instead of the large gate valves used in America.

There has been recently introduced in England a form of gate valve known as the Hopkinson-Ferranti valve, which has received wide application and is shown in Fig. 3. It embodies the well known principle of the venturi meter. The velocities are increased in the central portion of the valve so that the size is reduced to one-half the regular pipe size, while the contour of the passage on the delivery side reduces the velocity and restores the pressure. The flat valve faces are also reduced in size; hence are less liable to distortion. As the valve seats are small, this valve may be used to throttle the steam when opening up, rendering unnecessary the bypass valves common to the large gate valves. When the valve is opened it draws up a throat piece which when full open forms a continuous venturi tube and prevents eddying in the throat and dirt from getting under the seat. This class of valve can be used on steam lines where the velocities do not exceed 6000 feet per minute, but it is questionable if it could be used on steam lines to turbines where extremely high steam velocities have been obtained.

BOILER-FEED PUMPS

In Germany and in other parts of the Continent turbine-driven centrifugal boiler-feed pumps have been adopted extensively; turbines of the Curtis or the Electra type being generally used to drive them. One variation from American practice is that instead of exhausting these small turbines into feed-water heaters, which are seldom used, the exhaust is carried to one of the lower stages of the main steam turbines; otherwise they are run condensing. In England the vertical boiler-feed pumps of the Weir type are used almost exclusively, and with very satisfactory results. Improvement has been made on these pumps by the use of thin sheet-metal valves for both water suction and discharge. These valves resemble closely those used on the Leyner air compressor and the new Mesta blowing engines, manufactured in this country. The writer was informed that these can be operated at very high speeds and that under ordinary conditions they not only hold tight but last for some time if properly faced before being put into the pump.

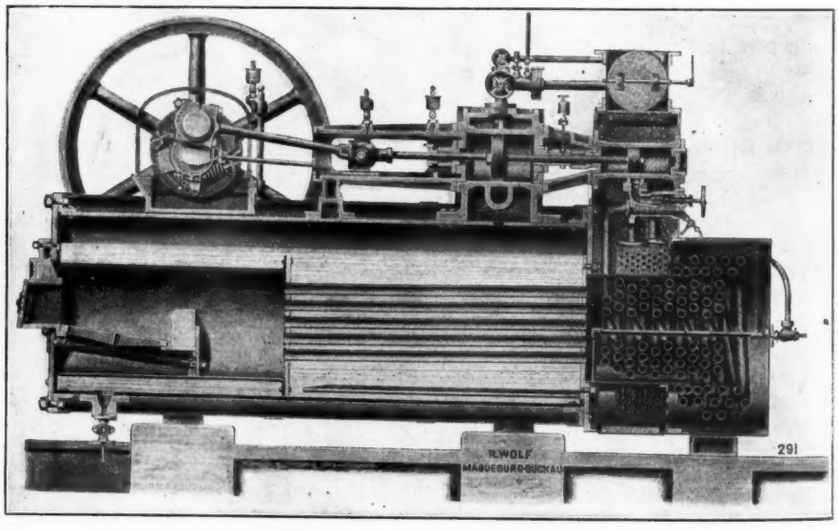


FIG. 5. WOLF LOCOMOBILE

Economizers

Economizers are used almost exclusively in Germany in connection with all classes of boilers and are also used very largely in England. In the latter country tubular boilers of the Galloway and Lancashire types are still largely employed, and where such boilers are installed the economizers are a decided advantage. Very few feed-water heaters were noticed.

CO_2 Recorders

These are to be found in almost all the Continental boiler rooms, and the firemen seem to watch them very closely and interpret the results shown. The type used is an improvement over those which have been introduced in America and apparently needs less attention. The firemen, however, have a second check placed on them by the installation of a recording instrument which shows the condition of the draft in the fire and in the chimney breeching. This combined with the CO_2 recorder gives a better record of the quality of firing than if the recorder alone were used, for most any degree of CO_2 can be obtained by choking the draft and working the boiler light. Recording gages are also installed for recording the boiler pressure.

Water Softeners

Water softeners are extensively used in all large European power plants. The engineers believe that it is more economical to remove scale by some mechanical or chemical means before it is put into the boiler than to remove it from the tubes after it has had a chance to settle.

Chimneys

In England one soon becomes familiar with the tall brick chimney which seems to pervade the whole manufacturing section of that country. In Germany these are also used extensively, but they differ in appearance from the English stacks as water towers are usually placed about half way up the chimney.

In Germany a new system of mechanical draft has been largely adopted by the Allgemeine Elektricitäts Gesellschaft and other contractors for power plants developing electricity where reliability is absolutely essential. This system has been patented by Dr. Hans Cruse, and one of its applications is shown in Fig. 4. It consists of an ejector-formed stack 40 or 50 feet high. Draft is obtained by blowing cold air through a pipe which discharges upward at the throat of the chimney, thus forming an air ejector. This air is supplied by means of a motor- or turbine-driven fan. As the air passes up the chimney to the throat it becomes heated and expands, thus increasing its velocity.

The advantages claimed for this system are as follows: The fan handles only cold fresh air; hence is not subject to deterioration from the destructive flue gases. It can be of efficient design and can be located in a place accessible for cleaning and for quick repairs. It does not require water-cooled bearings. It has to handle only small volumes of air at comparatively high pressures and therefore can be run at high speeds by standard and variable-speed motors. The draft can be regulated by varying the motor speed. The power consumption amounts to about 1 per cent. of the fuel burned: The cost of installation is about the same as for induced-draft outfits, and the apparatus would seem to be well adapted to furnaces where coals high in sulphur are burned.

TABLE 2. WOLF LOCOMOBILES, FROM OFFICIAL TESTS

Tested by	Date	Steam Pressure	Superheat, Degrees F.	Brake Horsepower	Pounds Steam per Brake Horsepower-hour	Pounds of Coal at 13,500 B.t.u. per Pound per B.H.P. Hour
Patented Superheated Compound Locomobiles with Condenser						
Magdeburg Boiler Association.	Dec. 2, 1904		200	288.9	10.8	1.32
Professor Gutermuth	Jan. 11, 1904		200	203.5	11.43	1.345
Professor Lewicki	Apr. 26, 1901	176 gage	250	108.5	11.66	1.37
Magdeburg Boiler Association.	Nov. 29, 1904		210	68.2	11.43	1.484
Superheated Compound Locomobiles, Noncondensing						
Magdeburg Boiler Association.	July 20, 1905		250	188.6	14.83	1.828
Magdeburg Boiler Association.	May 23, 1903		200	79.4	16.5	1.946
Superheated Tandem Locomobiles with Double Superheater and Condenser						
Royal Inspection Office, Magdeburg	July 20, 1905		230	46.7	11.03	1.376
			235	67	10.87	1.373
Professor Josse, Charlottenburg	June 3, 1904	180 gage	274	43.2	10.9	1.32
		180 gage	310	55.1	10.29	1.23
R. Mathot, Brussels		172.6 gage	275	34.7	10.8	1.33

The Locomobile

The great economy accompanying the use of highly superheated steam was early recognized by the Germans and this led to the development of prime movers of a type different from those used in this country. One of these types which continues to grow in favor is called the locomobile. Fig. 5 shows a section of a Wolf locomobile of the latest type, which consists of a boiler fitted with an internal furnace and short fire tubes, although for certain grades of fuel an external furnace of the dutch-oven type is recommended. This boiler is so constructed that the furnace tube sheets at both ends and the nest of tubes between them can be withdrawn for cleaning by breaking the bolted joints at the front and rear heads. This construction might be objected to in this country, but it has given no trouble in Europe. Steam pressures of from 150 to 200 pounds per square inch are usually carried in these units so that large ratios of expansion are possible. The outside of the boiler is heavily lagged. The engine is of the compound type and is placed on a saddle riveted to the top of the boiler shell. The high-pressure cylinder is in the smoke flue and is jacketed by the hot gases. The low-pressure cylinder is jacketed by the steam in the boiler and this jacket space acts as a steam dome. The steam is led from this portion through suitable piping to the first superheater, directly in front of the boiler tubes, and as these tubes are short the gases leave at a high temperature, making a high degree of superheat possible. This superheated steam passes directly through piston valves, controlled by an automatic governor, into the high-pressure cylinder.

Jacketing the cylinder by hot gases and using superheated steam reduce the cylinder condensation to a minimum; hence the steam is used very economically in this cylinder. The exhaust from the high-pressure cylinder passes into a second superheater, which is heated by the gases after having passed through the first superheater. Further economy is effected by providing a condenser, the air pump of which is usually driven direct from the engine shaft by an eccentric or, on large sizes, by a motor. The wasteful steam-driven boiler-feed pump has been replaced by a small pump, also driven from the main engine shaft by an eccentric. An injector is provided as a relay. The feed water passes through a feed-water heater placed in the exhaust pipe between the low-pressure cylinder and the condenser. Many of the later units have electric generators direct connected to the engine shaft. Platforms are provided around the engine and all parts are easily accessible for oiling, cleaning and repairs.

These units have shown remarkable economy, due to the following factors: A reduction of air leaks in the setting by having the furnace and superheaters

inclosed in the boiler shell; the use of highly superheated steam in both cylinders; the adoption of the compound engine with condensers and the use of high-pressure steam; the efficient jacketing of the cylinders; the elimination of almost all steam-pipe lines, and an efficient boiler-feed pump arrangement combined with heating of the feed. The great saving in floor space should also be noted.

The writer recently made inquiry regarding one of these units for laboratory work and received a proposal from R. Wolf, of Magdeburg-Buckau, Germany. The estimate was for one Wolf patent superheated condensing tandem locomobile with two superheaters and a boiler pressure of 176 pounds gage, the engines were to have balanced piston valves with a shaft governor, and a speed of 220 revolutions per minute. The capacities were to be: Economical load, 28 brake horsepower; maximum load, 45 brake horsepower; temporary overload, 54 brake horsepower. The engine was to be provided with a belt wheel. The guarantee was as follows: Consumption of coal between economical and maximum load not to exceed 1.52 pounds per brake horsepower per hour with coal having a heat value of 13,500 B.t.u. per pound and less than 5 per cent. refuse; the steam consumption not to exceed 11.66 pounds per brake horsepower per hour. Delivery guaranteed within three weeks of order.

This would correspond to a unit of about 40 indicated horsepower in America, and the guarantee compares favorably with the results obtained on large turbine units. When the load factor is bad and where absolute reliability and constant speed are essential, such installations are much superior to suction-producer plants. The price quoted was, roughly, $2750 f.o.b. Hamburg and packed for foreign shipment.

Other forms of these locomobiles are provided with compound cylinders which are jacketed by the steam in the steam domes instead of the gases leaving the chimney.

Recent results of tests on Lanz locomobiles at the Brussels exhibition showed with one unit of 178 brake horsepower, a coal consumption of 1.33 pounds per brake horsepower per hour, and a steam consumption of 11.69 pounds. On a larger unit of 235 brake horsepower the coal consumption was 1.286 pounds per brake horsepower per hour and the steam consumption 7.068 pounds. These results confirm those given in Table 2, which was made up from tests by a number of authorities in Europe.

Steam Engine Lubrication*

By R. D. Tomlinson

Various systems for external lubrication are discussed. Cylinder lubrication trouble is often remedied by changing the point of feed. Price is no guide to suitability of a lubricant. Wet steam is a source of trouble.

*Abstract of paper read at annual meeting of the Institute of Operating Engineers, New York, September 2, 1911.

During the last seven years the writer has been connected with a company which builds engines and has come in contact with many troubles due entirely to poor lubrication. In numerous cases the trouble was not due to the price paid for the lubricant or the quantity used, but to the fact that it was not suitable for the conditions of the particular engine on which it was to be used.

The lubrication of the pins and journals of the engine can best be accomplished, perhaps, by installing a gravity oiling system, for there has lately been put on the market an oil filter and pump which can be placed near the valve-gear side of the engine and driven from the rocker arm. This arrangement pumps the oil from the receiver below the floor up into the filter; from the filter the oil is pumped into a standpipe on the frame of the engine which gives sufficient head to force the oil to the bearings and pins. For small- and medium-sized units, this makes a satisfactory arrangement, as the entire apparatus is so located that the operator can see what is going on without leaving the engine room.

There are some locations where the old gravity system may be desirable. The main point that the writer wishes to convey is that some kind of a system should be installed whereby the bearings and pins of the engine can be practically flooded with oil; by so doing hot pins and journals can be eliminated. The engine will run two or three times longer without keying up (which proves that the pins and journals are not wearing rapidly) than would be the case if cups were used and drops of oil were fed to the pins and journals.

Another point in favor of the flooding oiling system is the fact that if from any cause a little dirt should get in the bearings or on the pins, the large quantity of oil passing through would wash the dirt out without doing any harm. On the other hand, if only drops are used, the probabilities are that the pins and journals would run hot and cause more or less serious damage if the attendant did not happen to notice them in time to take care of them. Hot pins or journals, like a fire, will soon get beyond control if not discovered in time.

There are many varieties of oil filter, some of which operate better than others, but the main point to look out for is the size of the filter. It should be large enough to allow sufficient time for the oil to filter properly while passing through.

Cylinder Lubrication

Cylinder lubrication is one of the most difficult problems with which the engineer has to deal.

A great majority of cylinder and piston troubles are due to improper lubrication, frequently because the cylinder oil used is not suitable to the conditions.

Variation in the temperature of the steam is an important factor, and a still more important one is the quality of the steam entering the cylinder. There is a great deal less trouble with the high-pressure cylinder than with the low as the steam is drier. The majority of cylinder oils will not properly perform their function if the steam is wet. Trouble from wet steam is much aggravated where large engines run for any length of time on light loads. The low-pressure cylinder steam is very wet and unless the cylinder oil used is compounded heavily with animal oils it will not form a film on the cylinder walls. Price is not a good guide to go by, for it frequently happens that a very low-priced oil, but one which is properly compounded for the service, will absolutely cure troubles which high-priced oils utterly fail to affect.

It is sometimes difficult to get a special oil, for many of the oil companies insist on supplying the regular oils, which they manufacture in large quantities. They seem disinclined to compound special oils to meet special conditions. There are exceptions, however, and satisfactory oils can be obtained if the user goes about it in the right way. A cylinder that shows evidence of improper lubrication, should be treated much the same way as any sickness; a competent engineer should diagnose the case and prescribe the proper oil to meet the conditions.

It frequently happens that the trouble is not with the oil itself but with the method of introducing the oil or the place where it is introduced. There are cases in which the oil pump or lubricator attachment was changed to some point on the steam line—sometimes 4 or 5 feet from the cylinder—with good results. This gives an opportunity for the oil to become thoroughly mixed with the steam

before entering the cylinder. There are other cases in which the oil should be introduced close to the cylinder; it all depends on the conditions. An expert for one of the largest oil companies in the country told the writer that when called in on lubricating troubles he often meets with success by changing the manner of introducing the oil into the steam and the place of its introduction.

The whole matter simply reduces to a careful study of the conditions that exist and experimentation with the various kinds of oil and the methods of application until the proper conditions are secured. This, any intelligent engineer can do.

A great deal has been written and said about the necessity of just the proper quality of iron in a cylinder to make a piston run properly in it, and there is no question of the advantages of the iron in the cylinder and bull ring being of the proper degree of hardness, etc. However, the writer has seen many soft-iron cylinders operate successfully without rapid wear with the use of properly selected cylinder oil that maintained a film on the surface of the cylinder.

One point in connection with the subject of cylinder lubrication to which I wish to call the attention of all engineers, is the fact that it is quite common practice to reduce the quantity of cylinder oil used to the least possible amount necessary. This cuts down the oil bill, but in many cases it increases the internal friction of the engine, so that it requires much more steam to turn it and, hence, increases the fuel bill. Engineers should remember that it is not the reduction of the oil bill or any other one item in a plant that counts; it is the reduction of the total cost of operating the plant that they should strive for. It is an easy matter to economize on some single item and by so doing raise the cost of operating the entire plant. Many an engineer, on taking charge of a plant, in his desire to make a showing reduces his help and supplies to an extent that shows up fine for the first few months. But a little later on he has found that the entire plant had to be overhauled and a large sum of money expended to bring the plant up to its original standard of efficiency.

Removing Emulsified Oil from Water*

By Darrow Sage

A small amount of ferric alum added to the water causes the oil to coagulate so that it is easily removed in a sand-filled filter.

*Abstract of paper read at annual meeting of the Institute of Operating Engineers, New York, September 2, 1911.

Unless one has had personal experience with the difficulties met with in removing emulsified oil from condensed water, he can hardly appreciate the persistency with which the oil sticks to the water. Water which in the form of steam has passed through the cylinders and steam valves of engines and pumps lubricated with cylinder oil, carries with it all the oil that has passed into the cylinders, some of it in the form of drops or small globules of pure oil and the remainder in the form of emulsified or finely divided oil in suspension. The emulsified oil gives the water a milky appearance similar in appearance to water that contains a quantity of air in suspension. In the case of emulsification this milky appearance does not leave the water, as when it is filled with air, and the oil cannot be filtered out by ordinary methods. This finely divided oil must not be confused with the drops of oil above mentioned, which are often seen floating on the surface of water condensed from exhaust steam, as these drops of oil may easily be removed from the water by collecting the mixture in a settling tank from which the water still carrying the emulsified oil can be drawn off at the bottom.

The removal of the emulsified oil to render the water suitable for re-use in the boilers is the real problem. At present there are three well known methods of removing this emulsion, namely:

(a) Filtration through certain kinds of natural sand or crushed stone.
(b) Electrical treatment.
(c) Chemical treatment.

Each of these methods has its particular field, depending upon local conditions and plant equipment. Only the chemical method will be described herein.

The commercial name of the chemical used in this process is ferric alum (ferric ammonium sulphate). This ferric alum comes in the form of lumps or crystals having a peculiar bluish tinge which turns to a brownish-yellow color after exposure to the air. It is easily soluble in water and this property is used to introduce the proper quantity into the water to be purified. From experience it has been learned that about 4 pounds of ferric alum will successfully clarify 10,000 cubic feet of emulsified oil and water, and if this water is recovered at the rate of, say, 4200 cubic feet per hour, the ferric alum should be added uniformly at a proportional rate. Not only must the rate of flow of the water and ferric alum be uniform, but care must be taken to see that the chemical is thoroughly mixed with the water to be treated.

A convenient way of doing this is to dissolve the necessary amount of ferric alum to last about 24 hours in a barrel of water and permit this solution to flow or drip into the tank in which the return water is caught before it is pumped to the filter, as will be explained later. In making the ferric-alum solution, care must be taken not to produce too strong a mixture, as a saturated solution of this chemical in water is very corrosive. It should never be mixed in greater proportions than 2 pounds to the barrel of water of about 50 gallons and where conditions will permit, less than this. As soon as the ferric-alum solution and the water under treatment have been thoroughly mixed, a pronounced change can be noticed in the milky appearance of the water. It is caused by the action of the ferric alum, which coagulates the minute particles of oil suspended in the water, and thus produces a change that cannot be easily described but can be quickly detected.

After the action of the chemical has taken place the mixture is ready for filtration.

A filter for this purpose can be made from an ordinary tank partially filled with coarse, clean sand which has sufficient area to permit a flow not exceeding 15 cubic feet per hour per square foot of sand bed. The sand bed should be about 2 or 3 feet thick and the sides of the tank must extend above the top of the bed a sufficient distance to provide a separating space for the floating oil. Into this filter the water treated with the ferric-alum solution should be pumped and properly scattered about the surface of the sand bed so as not to wash holes therein and be allowed to pass out at the bottom of the bed through screens and perforated pipes to keep the sand back. The emulsified oil being properly coagulated, it will be caught by the sand and from the discharge of the filter clear water practically free from oil will be obtained. Where it is possible to arrange the filter so that the overflow can be caught, the drops of oil and coagulated scum can be washed out of the filter by simply shutting the discharge valve from the bottom of the tank and permitting the filter to overflow its sides. The floating oil can in this way be skimmed from the top of the tank at regular intervals, but its use as a lubricant is not advisable. By this method the cleaning of the sand bed can be materially postponed, thus getting the equivalent of a longer life out of the filter. With a filter as above described, cleaning will be necessary about every 30 days when working up to its full capacity, it being assumed

that there is no more than a normal quantity of oil in the water which passes through the filter.

The cost of operation of a filter plant of this size is small compared to the value of water saved, as the present price of ferric alum is about 20 cents per pound and the cost of saving 10,000 cubic feet of water would be 80 cents plus the cost of the pumping and whatever slight attendance that is necessary to operate the pump and keep it in good condition.

Turbines for Japanese Navy

It is proposed to adopt for the Japanese navy 25,000 brake horsepower of Curtis turbines for two of the first-class battleships, "Kawachi" and "Settsu," and 22,000 brake horsepower of Curtis turbines for two out of three scout cruisers. For the remaining scout, one set of Parsons turbines of the same power will be used and two sets of the Parsons type of 20,500 horsepower for two destroyers. The latest improved Parsons turbines of 64,000 brake horsepower are also to be installed for three first-class armored cruisers, one to be built in England and the remainder at home, and a Curtis turbine installation of the same horsepower is to be placed in another first-class armored cruiser, which will be built at home.

The accompanying engraving, reproduced from *The Engineer*, shows the 15-stage Curtis turbine for the battleships "Kawachi" and "Settsu." The diameter of the turbine is 12 feet at its pitch circle, and it is designed for 27,000 brake horsepower at 270 revolutions for the 7-stage and 25,000 brake horsepower at 245 revolutions for the 15-stage turbine. The latter consists of six ahead wheels and one ahead drum, carrying nine stages and two reverse wheels. Reaction blading has been used on a part of the drum and second-stage astern wheel of all the Curtis turbines to counterbalance the propeller thrust by means of steam thrust, so that the thrust shaft and collar may be made comparatively short.

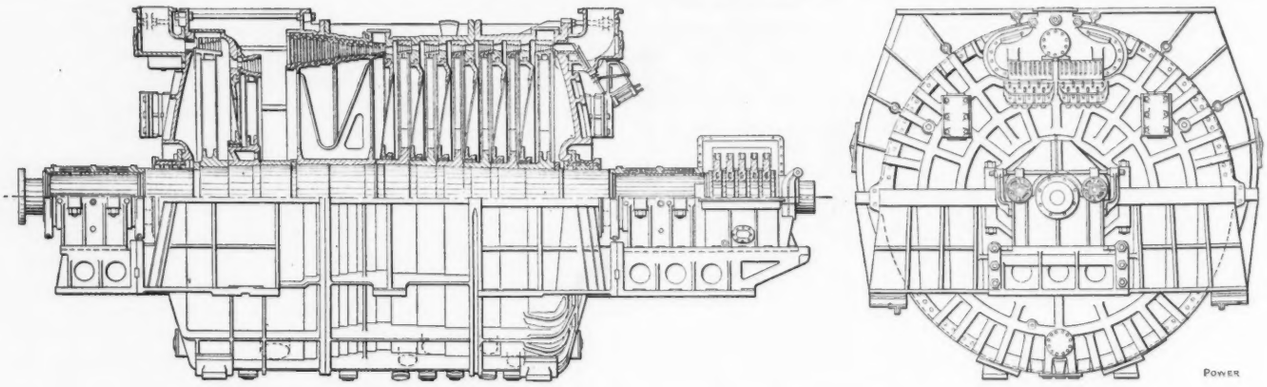
Special Crane for Coal

Coal handling at the plant of the Toledo Railway and Light Company, at Toledo, O., presented a peculiar local condition which was solved by the combined gantry crane with the cantilevered coal elevator shown in the accompanying illustration. This plant is located on the bank of the Maumee river, and the two main-line tracks of the railroad are alongside of the power house, between it and the side tracks upon which the coal cars must be placed.

To avoid tunneling below the tracks an overhead handling system was employed which would permit unloading cars at any point on the siding and the transfer of their contents to the storage bunker in the power house.

There was designed for this purpose a gantry electric-locomotive crane with a boom of 40 feet radius, which is sufficient to unload three gondolas without moving the gantry. The crane also handles coal from barges or canal boats lying at the wharf. It is mounted over the wharf leg of the gantry and operates a one-ton clamshell bucket which dumps into a 20-ton hopper carried by the gantry. From this hopper the coal flows to the conveyer boot, from which it is elevated to a discharge spout which can be spotted to fill any desired section of the bunker by moving the gantry. While the capacity of the crane is about 80 tons per hour and the conveyer 160 tons per hour, the device is operated by one man located in the cab of the crane where all the controllers and operating levers are centralized. The cost of handling coal with this machine is about 2.5 cents per ton.

COAL-HANDLING CRANE OF TOLEDO RAILWAY AND LIGHT COMPANY



CURTIS TURBINE FOR NEW JAPANESE BATTLESHIPS

A High Speed Cast Iron Flywheel

By W. Trinks

A 17-foot wheel designed for a rim speed of 10,000 feet per minute. It has no rim joints and the spokes are cast solid with the rim, but free at the hub ends. Weakness due to flaws was avoided by especially careful foundry work.

Average practice in the design of cast-iron flywheels dictates an upper limit of a mile a minute or 88 feet per second for the mean rim speed. Since the hoop stress in a revolving cast-iron ring is given by

$$\frac{(\textit{feet per second})^2}{10}$$

the hoop stress in average flywheel practice is less than 800 pounds per square inch. The tensile strength of cast iron is about 22,000 pounds per square inch and at first thought the factor of safety of

$$\frac{22{,}000}{800} = 27.5$$

seems ridiculous. Yet the occasional bursting of a flywheel preaches caution and deters engineers from adopting higher speeds. It is known that the difference between simple theory and actual practice is caused by four different agencies:

(1) The weakening of the rim by the joints of sectional wheels;

(2) Casting stresses;

(3) The bending of the rim by the forces in the arms;

(4) Flaws.

It is found by even approximate calculation that the weakening effect of the common form of joint is very great indeed. The author has never seen a complete theory of flywheel joints which considers the concentration of local stress due to the action of the fasteners (links, bolts or keys), the influence of the bending moment and the action of shrink links due to their initial tension beyond the elastic limit.

Casting stresses depend not only upon the design of the wheel but upon foundry practice as well; they are uncertain, and in many designs of wheel their presence cannot be detected. The author remembers a wheel of standard design which, while lying peacefully in the yard of a foundry, pulled an arm off, near the hub, even before the wheel had been delivered to the machine shop. To judge from this one example, even no speed is too high for a flywheel. Fortunately, such a sad combination between poor design and poor foundry practice occurs but seldom.

The bending of the rim can produce unexpectedly high stresses, particularly in shallow-rim pulleys. These can be mathematically determined, however, and thus kept within safe limits. An example will be given later.

Flaws furnish another valid reason for using a high factor of safety. They are similar to shrinkage stresses inasmuch as they escape discovery so long as the wheel is intact. They differ from shrinkage stresses because flaws are usually located by a post-mortem examination. Sudden changes of cross-section favor the formation of flaws and should therefore be avoided. A discussion of the manner in which to avoid flaws by correct foundry practice does not belong in this article.

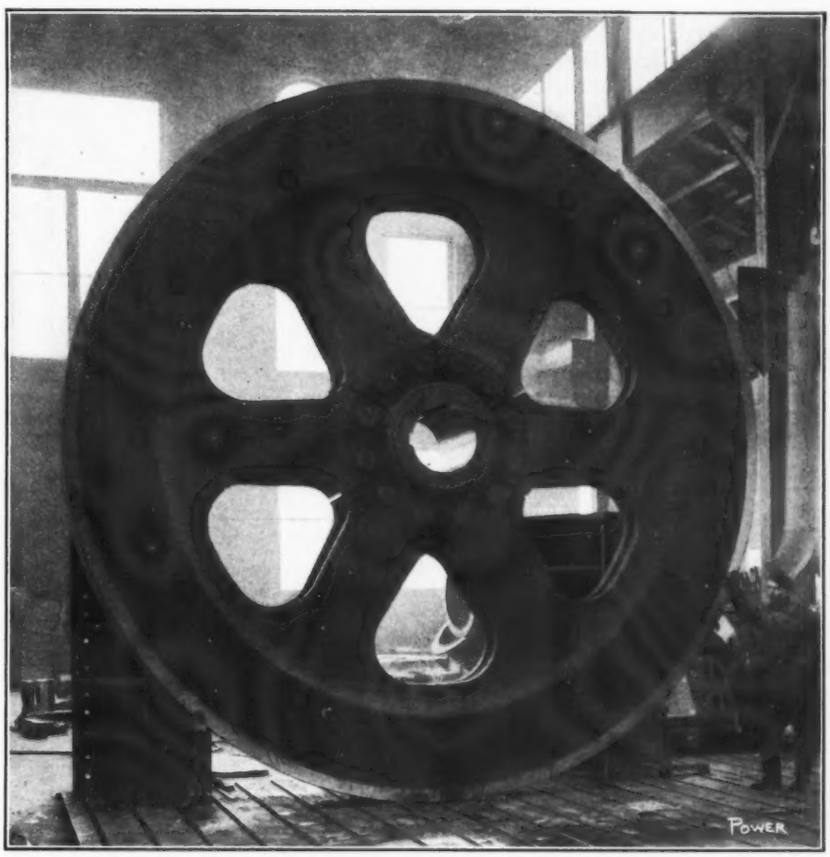


FIG. 1. FLYWHEEL DESIGNED FOR A RIM SPEED OF 10,000 FEET PER MINUTE

In the summer of 1910, the author designed for the Mesta Machine Company, builder, and the Cambria Steel Company, purchaser, a flywheel with an outside-rim speed of 10,000 feet per minute.

The wheel is shown in Figs. 1 and 2. It will be seen that there are no rim joints. It was found that a 17-foot wheel could be shipped solid from the works of the builder to those of the user in spite of the distance, about 65 miles. The absence of rim joints, of course, eliminates the weakening effect of such joints.

The elimination of casting stresses was quite serious. It is well known that, in general, large solid-rim wheels are shunned by engineers, not only on account of the difficulties of transportation and erection but because great and uncertain shrinkage stresses are found in them. The illustrations show the various features which were employed to avoid these stresses. First, the arms, while made solid with the rim, were not connected at the hub; thus, they could shrink independently of each other. Second, the arms pass with long sweeping curves into the rim and into the hub section for the purpose of shoving the sand of the mold sideways by wedge action and thereby avoiding surfaces which could grip the sand and thus prevent free and unhindered shrinkage. This feature was made possible by splitting the wheel into two wheels, bolted together at the rim and connected to a common hub. Third, the dishing of the arms allows bulging out during the cooling of

the casting, if resistance should be offered by the central core. It may be mentioned here that the foundry did its share in avoiding stresses, for instance, by making the cores between the arms soft and comparatively loose so that they could not offer undue resistance to shrinking.

The fourth weakening effect, namely, flaws, may be mentioned in this connection. Splitting the wheel in two was done with the intention of avoiding unduly heavy sections. "Soundness to the core" is much harder to obtain in heavy sections than in light or medium sections, as explained in the various textbooks and periodicals on foundry practice. Besides, the gradual passage from the arm section to the rim section tends to prevent cavities in the casting.

With the shrinkage strains practically eliminated, the determination of centrifugal strains in the arms and in the rim can be made with some accuracy. Since a woeful amount of ignorance exists on the distribution of forces in flywheels, a review of the method of calculation will probably be welcomed by "seekers for the truth."

Imagine the rim of a wheel rotating by itself—that is, disconnected from the arms; then its diameter will grow a small amount. Next imagine the arms rotating disconnected from the rim; then they will also grow in length, but not as much as did the rim. If both rotated separately, there would consequently be a gap between the arms and the rim. In reality there is no such gap, from which it must be concluded that the rim pulls the arms out and that the arms pull the rim in. From the size of the gap and the dimensions of the wheel the force acting between the rim and the arm can be computed in this manner. The arm is a rather stiff spring with a certain scale; the rim is a spring with another scale; the imaginary gap furnishes the deformation of both springs due to the force under consideration. These data together with the self-evident condition that the force on both springs must be alike (one being the reaction for the other) make the calculation determinate. A few words may be said about treating the arms and the rim as springs. The arms are straight bars and their deformation by direct tension is easily computed. The rim presents more difficulties as its deformation is made up of two components: First, a reduction of the diameter of the rim by the hoop tension which balances the arm pull; second, a flexure of the rim by the arm pull. All of these calculations; namely, the computation of the size of the gap, the stiffness of the two springs, the rim-to-arm pull and the resulting stresses, are thus purely a matter of applied mechanics; they offer no fundamental difficulties, but are very tedious, because no general formula can be employed.*

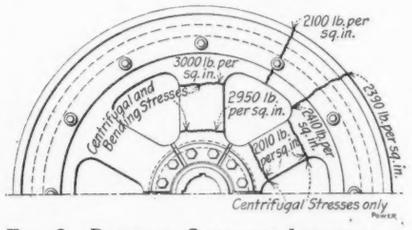

2100 lb. per sq. in.
3000 lb. per sq. in.
Centrifugal and Bending Stresses
2950 lb. per sq. in.
2390 lb. per sq. in.
2010 lb. per sq. in.
Centrifugal Stresses only
POWER

FIG. 3. DIAGRAM SHOWING LOCATION OF STRESSES

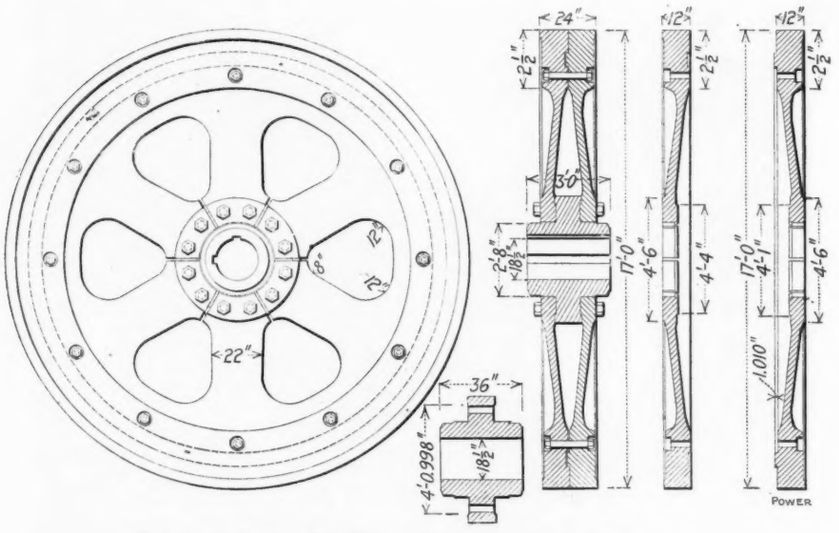


FIG. 2. SHOWING PRINCIPAL DIMENSIONS OF FLYWHEEL

Undoubtedly an outer-rim speed of 10,000 feet per minute will be called recklessness by many, and for their special benefit the stresses found by the above outlined method will be given.

The principal stresses have been entered on the diagram in Fig. 3. The ideal hoop stress, as computed from the formula for a thin revolving ring, amounts to 2150 pounds per square inch. On account of the depth of the rim the bending by the arm pull does not increase this stress materially. Thus the stress in the rim near the arm is 2410 pounds per square inch, whereas the stress midway between the arms is only 2100 pounds per square inch. The stresses in the arms due to centrifugal force are also small, amounting to 2390 pounds per square inch near the rim and to 2010 pounds per square inch near the hub. The material of the wheel (air-furnace iron) showed a tensile strength of 30,000 pounds per square inch. Since the design practically eliminates all casting stresses, this wheel should be very safe in spite of its speed.

Naturally, flywheels are intended to store up and give out energy, which process involves bending stresses in the arms. In the wheel under discussion the power to be transmitted was not specified by the purchaser because the varying temperature of steel in the rolls will vary the power required between extremely wide limits. Therefore, the builders reversed the method of procedure and computed the greatest power which can be transmitted by the wheel at a given speed with a given fiber stress in the arms. If the sum of centrifugal and bending stresses is limited to 3000 pounds per square inch in the arms, as indicated in Fig. 3, the wheel can deliver or store up power at the rate of 21,000 horsepower. The builders felt satisfied that the rolls or roll housings would let go before the wheel if any such amount of power were required.

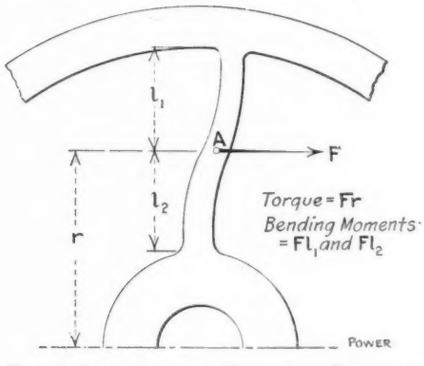


FIG. 4. ILLUSTRATING EFFECT OF BENDING STRESSES

A few words should be said on the subject of bending stresses in the arms because much confusion exists in the minds of many engineers on this point. Each arm is rigidly held at the hub, and is more or less rigidly held at the rim; it can therefore deflect only as is shown in Fig. 4, presenting an inflexion point *A* whose location depends upon the relative stiffness of the arm and the rim. In flywheels proper with deep rims the point *A* lies near the middle of the arm. In

*Stanwood and Lanza tried to derive such formulas (see *Transactions* of A. S. M. E. of the years 1892 to 1895). J. Goebel gave a complete derivation in Z. D. V. D. I. (March, 1898), but his formulas, although correct, are too complicated for practical use.

pulleys with shallow rims the point *A* lies near the outer end of the arm. The exact location of the inflexion point can be mathematically determined, but the process is tedious, and a skilled designer soon learns to guess the location of this point with considerable accuracy. Force *F*, Fig. 4, times the radius *r* gives the transmitted moment. This simple reasoning teaches that the bending at the rim is frequently just as great as the bending near the hub.

The outer rim speed of this wheel, as before mentioned, is 10,000 feet per minute, but this is by no means the commercially attainable limit. Wheels with plate walls gripping over a steel rim in halves can be run safely at 15,000 feet per minute. The simple formulas for hoop stress then are no longer applicable, because solid disks have taken the place of arms. We all know that properly designed disks can be run at very high speeds, the De Laval disk, for instance, running at a speed of 72,000 feet per minute.

While it is thus apparent that much higher rim speeds may be used than have heretofore been customary, it should not be overlooked that commercial rim speeds depend not only upon strength of materials and design but also very largely upon dollars and cents. Rotative speeds in steam-engine practice have been moderate, and low rotative speeds do not permit high rim speed without excessive diameters of wheels. An extreme case will illustrate the point: An engine which is direct-connected to a tin-plate mill and runs 30 revolutions per minute would have to be equipped with a 160-foot diameter wheel to attain the before mentioned rim speed of 15,000 feet per minute. Such a wheel would be all arms or all plate without any rim and would be preposterous. Electric motors and high-speed gas engines, on the other hand, can make use of high peripheral speeds. The proper rim speed for flywheels has, therefore, ceased to be solely a question of strength but must be decided from case to case.

Why Jimmy Was Refused a License

By J. E. Terman

James Thomas, a young man of 24, was a technical-school graduate, but having no financial resources he was compelled to start in at the bottom of the ladder. In the beginning he was making the armature repairs for a street-railway system in the South and was called upon to run the engines when either of the two engineers was absent. Here was Jimmy's opportunity to become familiar with steam engines; he had had practical experience with boiler operation.

The town in which the plant was located had a municipal license law for engineers, and, although Jimmy was aware of this fact, he never thought that the conditions under which he was operating the plant demanded a license. One day he received a notice to report to the board of examiners, with threats of dire penalties should he attempt to further operate the engines without a license. Jimmy, greatly disturbed by the receipt of this notice, consulted the chief engineer. The chief told him that all of the members of the board were mere starters and stoppers and that he need not fear any puzzling questions from them. Jimmy skimmed through a textbook on the steam engine to refresh his memory on some points of engine design and operation.

On the day appointed for the examination, Jimmy came before the board of examiners. The questions were all of a very practical nature, such as the correct location of the water line in a boiler, how to set a safety valve, etc., and Jimmy found no difficulty in answering them with entire satisfaction, until finally Shaffer, one of the throttle twisters, asked how the clearance of a steam engine could be determined. Jimmy felt very thankful that the professor at college who had taught him this subject had been thoroughly practical, and he remembered the simple method of measuring the clearance of a steam engine of the Corliss type when the valves and piston were tight (by placing the engine on the center at the end where it was desired to measure the clearance and pouring a measured quantity of water into the steam port until the clearance space was filled). Jimmy was sure that the simplicity of the thing would win the examiners so he described this method in answering the question, supplementing it with the information that all spaces might be measured by removing the cylinder head and in this way the clearance could be calculated.

Jimmy had progressed so well up to this point that he was quite taken back when Shaffer shook his head and said that his answer was wrong. Shaffer said that to correctly determine the clearance on any steam engine the crank should be placed on the center and the piston rod disconnected from the crosshead and then the piston moved until it struck the cylinder head; the space through which the piston moved would be the clearance.

Jimmy stated that this kind of clearance was commonly termed striking clearance, and that the questioner should have specified striking clearance if that was what was wanted; that if any engine builder was asked regarding the clearance of an engine, he would give the total amount of cubic space left behind the piston in the cylinder and in the ports, when the engine was placed on the center, and would express it as a percentage of the displacement of the piston in making a stroke.

Farrin sided with Shaffer, but Rashley, the third member, sided with Jimmy, who was treated to an edifying discussion on what was steam-engine clearance by the three members.

Jimmy had gained confidence by listening to the harangue of the board members over the subject of clearance, and boldly stated that he wanted a first-class license. Shaffer held up his hands in horror at the idea and said that if they should give a young electrician a first-class license, they could give no excuse when a man who had been passing coal and firing boilers for ten years should come to them and be granted only a third-class license.

Jimmy was firm, however, and told Shaffer that he did not "give a hoot" what kind of excuse they gave such a man; it was a first-class license or nothing with him, unless they could furnish some good excuse for witholding it. As the examination was the only basis allowed for determining the qualifications of an applicant for a license, and as Jimmy had answered all of the questions satisfactorily except the one that the board itself could not agree upon, he had very much the best of the argument.

Rashley at this point beckoned Jimmy to step out in the hall with him, where he told Jimmy that he knew the street-railway people well, and he did not want to inconvenience them by preventing Jimmy from operating the plant when necessary. He said that while he felt that Jimmy should have a first-class license on the strength of his examination, he wished that he would consent to accept a second class as the other two examiners were opposed to technical graduates, and he felt sure that they would never consent to issue a first-class license.

This matter of objecting to him on account of his education made Jimmy "hot," and he returned to the room and told the examiners that it was a first-class license or none. Shaffer said, "All right, young man, you will get none, and if you are caught operating that plant again you will be arrested and it will go hard with you." Jimmy well knew that, legally, the license law was about as strong as tissue paper, and he now had all of his Irish aroused, so he told Shaffer where he could go; that he proposed to operate the plant the first time occasion demanded it, and defied them to have him arrested. "I would like nothing better," said Jimmy, "than to get you three engineers up before a judge and jury to determine your fitness to make examinations for issuing dog licenses, much less determining a man's ability as to whether he can be intrusted with the management of boilers and engines."

Jimmy was never molested, although he ran the plant many times after he failed to get his license.

Heat Transmission in Boilers*

By V. L. Rupp

Some 1¼-inch pipes plugged at the ends were fitted into the 4-inch tubes of a horizontal tubular boiler. The results of the tests showed a 10 per cent. increase in the evaporation obtained per pound of coal.

*Abstract of paper read at annual meeting of the Institute of Operating Engineers, New York, September 1, 1911.

Having been convinced from observation and investigation that a 4-inch tube is too large to use in a horizontal tubular boiler 15 feet in length, and that a large part of the gases passes through the upper rows of tubes, I decided to make some experiments with a view to ascertaining what improvements could be effected.

The boiler on which the experiments were made was 15 feet in length, 5 feet in diameter, and had forty-four 4-inch tubes and was rated at 80 horsepower. The grate area was 25 square feet. A hole was drilled through the boiler front at *A*, Fig. 1, to take the temperatures of the gases after they had passed through the tubes.

Another hole was drilled at *B*, Fig. 1, through the side wall of the setting at the back of the boiler to take the temperature of the gases before entering the tubes.

After the boiler was put into regular service a series of temperature readings was taken. These readings are plotted in Fig. 2. The average temperature for the front of the boiler was 559 degrees Fahrenheit and for the back 1388 degrees. This shows a drop in temperature through the tubes of 829 degrees.

Since the volume of a gas at constant pressure varies directly as its absolute temperature it is well to note the difference in volume of the gases at points *A* and *B*, Fig. 1. The ratio is nearly 2 to 1. Now, if the volume of the gases at *B* is twice as great as at *A*, the gases will travel at twice the velocity at the back end of the tubes that they do at the front end. This being the case, the tubes are much more efficient in the conduction of heat at the back end than they are at the front end, if the theory that heat transfer is influenced by the gas velocity is valid.

I decided to try to increase the velocity of the gases through the entire tube length and at the same time bring the gases into more intimate contact with the heating surface. After some calculations regarding the amount of gases passed by the tubes and finding that I could cut down the tube area and still pass sufficient gas to burn the required amount of coal, I had a small casting made as shown in Fig. 3. This was made so as to be a neat fit in the tubes and allow a 1¼-inch pipe to pass through the center. A quantity of old 1¼-inch pipe which had been taken from the heating system was lying in the stock room. This I had cut into 15½-foot lengths. One casting was put on the pipe 12 inches from each end and one in the middle; they were fastened in place with small setscrews. The pipes were then sealed at each end with fireclay and put into the tubes. The boiler was not cut out of regular service after the pipes had been put into the tubes and everything was done to have the same conditions as when the previous temperature readings, shown in Fig. 2, were taken.

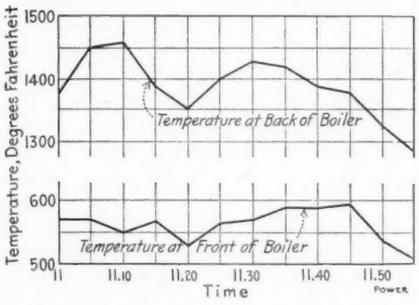


FIG. 2. TEMPERATURES BEFORE PIPES WERE PUT IN

A series of temperature readings was taken with the pipes in the tubes. The temperatures observed are plotted in Fig. 4. The average temperature at the back of the boiler was 1531 degrees and at the front, 486 degrees. This shows a drop through the tubes of 1045 degrees. It was found from a great number of readings that if the average temperature at the back of the boiler was over 1500 degrees, the temperature at the front without the pipes in the tubes was 625 degrees or over.

The boiler was forced with a blower to see how high a temperature could be obtained at the front end. Fig. 5 gives the results. The average temperature at the front of the boiler was 552 degrees and at the back 1686 degrees. This gives a drop through the tubes of 1134 degrees. One thing that should be noted in particular in this test is that after the temperature at the front reached 565 degrees it seemed to hang at that point, although one of the readings of the pyrometer at the back gave a temperature of 1785 degrees. It was not deemed advisable to force the boiler longer than 35 minutes as it began to prime.

I was not fully convinced that the improvement observed was due entirely to the 1¼-inch pipes in the tubes as it

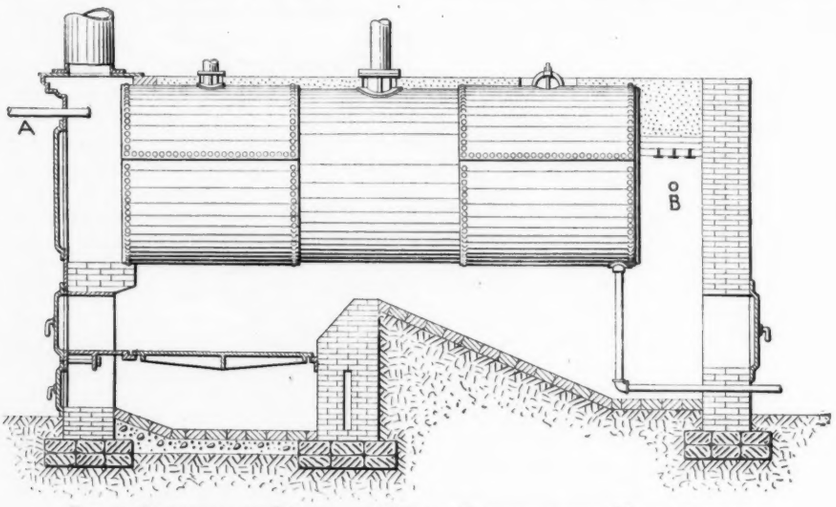


FIG. 1. LOCATION OF POINTS AT WHICH TEMPERATURES WERE OBSERVED

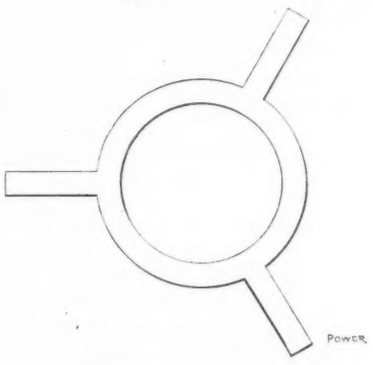


FIG. 3. CASTING FOR HOLDING PIPES IN TUBES

seemed possible that the restricted tube area caused a more equal flow of the gases through the tubes as a whole. Therefore, I removed the pipes and inserted a small metal ring in the end of each tube so as to give the same area for the flow of the gases as did the pipes.

While this showed some improvement over the conditions obtained with the boiler unchanged in any manner, the results were not to be compared with the improvement shown by the use of the pipes in the tubes. The improvement was great enough, however, to bear out the theory that the larger part of the gases passed through the upper rows of tubes.

Boiler tests were run with and without the pipes in the tubes. The test with the pipes in the tubes showed a 10 per cent. greater evaporation per pound of

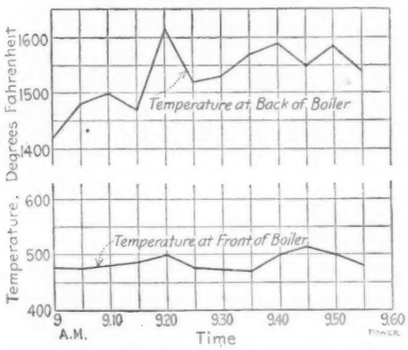


FIG. 4. TEMPERATURE READINGS WITH PIPES IN TUBES

coal. The boiler was not prepared in any way for these tests; it had been in regular service for about three months before the tests were made and was not opened until after they had been completed.

Fig. 6 gives two sets of temperature observations; the ones indicated by the dotted lines were made with the boiler equipped with the pipes and the others without. The sudden rise and fall of the curves are due to the damper being opened and closed, and to the coaling of the fires.

The pipes have been in the boiler for about two years and are in good condition with the exception of about a foot at the back end. The fuel used was buckwheat and no trouble was experienced in keeping the tubes clean and free of dust.

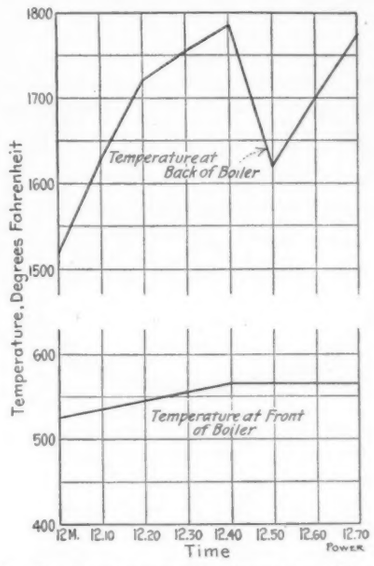


FIG. 5. TEMPERATURES OBTAINED WHEN FORCING BOILER

Wooden vs. Steel Boilers

An accident to the machinery of the steamer "Argyle" on her way from Toronto to Whitby, on Lake Ontario, recently disabled the vessel so that it was necessary to remove the passengers and transport them by rail to their destination. The following garbled report of the accident in one of Toronto's daily papers will no doubt be appreciated:

"The 'Argyle' is equipped with two wooden and two steel boilers, and the wooden boilers, which are the easiest on the rest of the machinery, had been used right through this season up to yesterday.

"Yesterday morning, however, the steel boilers, which cause much strain on the machinery, were installed and this was the cause of all the trouble.

"When the 'Argyle' started out the greater vibration of these proved too much of a strain on the rest of the machinery, the trouble being made more apparent owing to a wooden patch with which one of them had been repaired.

"It was the added vibration of these that jarred the valve pin out of place, and thus cutting the steam off, caused the machinery to stop and the side wheels to come to a standstill.

"Had the steamer been kept going with the valve pin out of place, the piston rods would have knocked in among the cylinders and very serious consequences would have resulted. As soon as the passengers were landed the six members of the engineering department of twelve, who were on duty, were at once put to the work of replacing the steel boilers with the wooden boilers.

"Before morning they expect to have made the change, and with a new valve pin replacing the one that was shaken out, the machinery will be in good working order again. With the wooden boilers in place it is stated that the machinery works in first-class shape, and no further trouble is looked forward to."

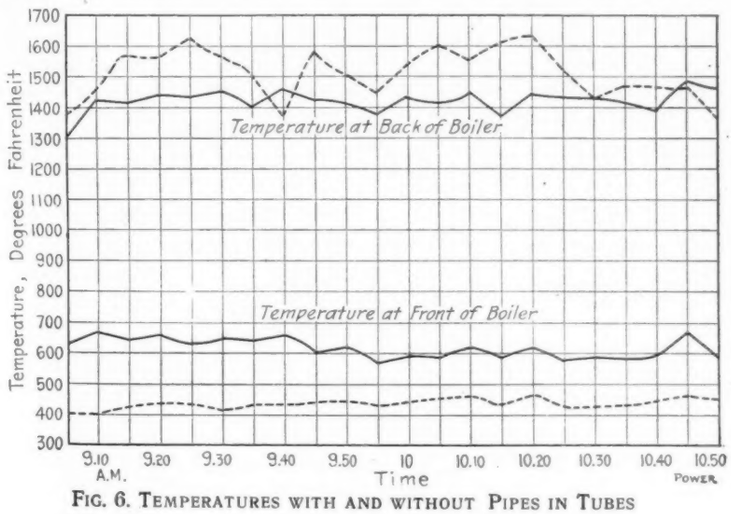
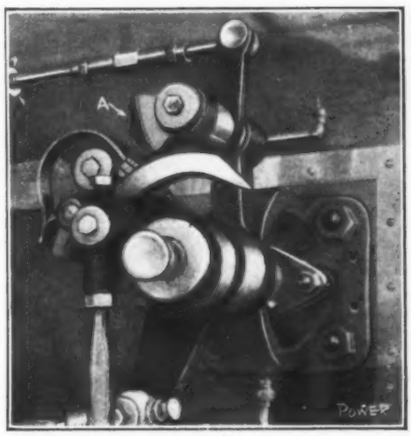


FIG. 6. TEMPERATURES WITH AND WITHOUT PIPES IN TUBES

Wooden Knockoff Plate

The accompanying illustration shows the Corliss valve gear on one end of a pumping engine. The knockoff cam *A* was originally fitted with a steel plate which, owing to the movement of the valve gear, caused a click just before the catch block hooked on to open the steam valve. While this clicking did no harm, it was exceedingly annoying.



WOODEN KNOCKOFF PLATE INSTEAD OF STEEL

In order to eliminate this noise, the engineer of the plant removed the steel knockoff plate, which was held in place by screws, and substituted one made of lignum-vitae. It did the trick, and shows practically no wear. The dark portion of the knockoff cam shows the new wooden plate.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men

Gas Engine Cycles

By Cecil P. Poole

A cycle, in engineering, is any series of operations that leaves conditions the same at the end that they were at the beginning. To illustrate this definition, consider the case of the water which is made into steam to drive a condensing engine. If the water is taken from a river, as is frequently the case, it passes through the following series of operations or "events," as they are usually called:

Being lifted from the river by a pump and forced into a boiler; evaporation into steam; expansion in the engine cylinder; cooling to liquid form in the condenser; discharge into the river. This series of events forms a cycle, because when the water gets back into the river it is exactly where and as it was at the beginning of the events.

A cycle, however, can be performed without reference to materials or substances. For example, a steam engine takes in steam, expands it, discharges (exhausts) most of it, compresses the remainder, and is then ready to take in more steam and repeat the events. This is a cycle also; because the engine is

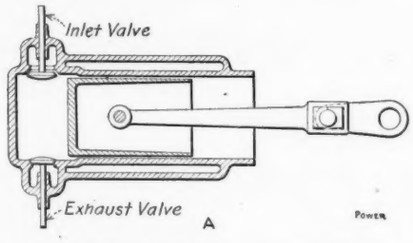


Fig. 1. Beginning of the Cycle

in the same condition after driving most of the expanded steam out and compressing what is left as it was before taking in the live steam, after it has once got into operation. The warming up and filling the clearance before starting are ignored because they are merely getting-ready processes and do not form part of the cycle, which is repeated over and over after the engine is once started.

The cycle of a gas engine is not so simple, because it includes both the generation and application of the driving force. That is, it takes in the working "fluid" (a mixture of gas and air), as it is called, when it is in a state of impotence—that is, it cannot do any work—and has to change the state of the "fluid" by burning it and thereby setting free the heat in it and producing a working pressure. The steam engine, on the contrary, receives its working "fluid" (steam) in a state of compression to a high pressure and only has to apply this pressure, which has been generated in the boiler. The gas-engine cycle consists of five events, as follows:

Admission, or suction;
Compression;
Combustion;
Expansion;
Exhaust, or expulsion.

These five events occur over and over, in the order named, no matter what kind of a gas or oil engine may be used. Different forms of engine perform the cycle by means of different kinds of mechanism, but the fundamental cycle is the same in all of them.

There are two general classes of gas and oil engines, grouped according to the relation between the carrying out of the cycle and the movements of the piston. Taking a simple single-acting engine for the basis of the explanation, one class requires four strokes of the piston in order to go through the complete cycle and the other class performs the complete cycle in two piston strokes. Engines of the former class, therefore, operate on the four-stroke cycle; engines of the latter class operate on the two-stroke cycle.

The Four-stroke Cycle

Remember that this discussion, for the present, refers only to engines which have trunk pistons working in single-headed cylinders—that is to say, a cylinder made with one end of the barrel closed by a head and the other end entirely open when the piston is not in it. This type of engine will be recognized readily upon looking at Figs. 1 to 6, although the pictures are largely diagrammatic.

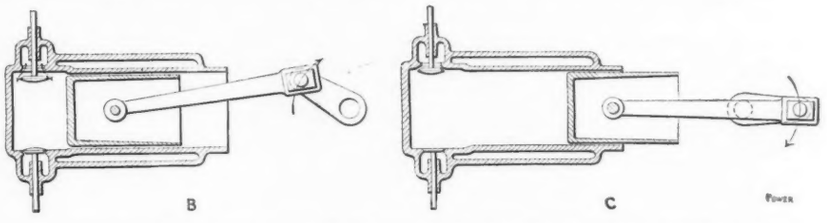


Fig. 2. The First Stroke (Suction); Drawing in the Charge

In this form of gas engine, working on the four-stroke cycle, one complete piston stroke is devoted to taking in a "charge" or cylinderful of a combustible mixture of gas and air; the return stroke compresses the charge; at the end of this stroke, while the crank is passing the dead center, combustion occurs, producing the rise of pressure necessary to give power to the engine; then an outward stroke is devoted to expansion, during which the gases push the piston forward and deliver power to the crank shaft, and the succeeding stroke clears the burned gases out to make way for the next charge. To illustrate the cycle, the familiar diagrams of the four strokes, somewhat amplified, are presented herewith.

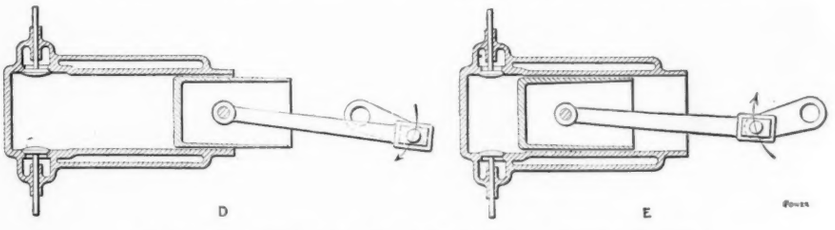


Fig. 3. The Second Stroke; Compressing the Charge

In Fig. 1 the crank is shown on the

inner dead center and the inlet valve is just beginning to open to admit the charge of gas and air, which begins to be drawn into the cylinder by the suction of the piston a moment later, as represented at *B*, Fig. 2, where the inlet valve is shown wide open and the entrance of the charge is indicated by the two arrows.

While the piston moves from the position *A* to that of *C*, the charge is drawn in, and this constitutes the "suction" or admission stroke; the inlet valve is shown just about to seat, after which compression begins, as illustrated at *D*, Fig. 3. (A comparison of the piston and crank positions at *C* and *D* will show how deceptive it is to consider gas-engine events from the viewpoint of crank motion. Although at *D* the crank has moved about 19 degrees from the dead-center position *C*, the piston has moved about one-fiftieth of its stroke.)

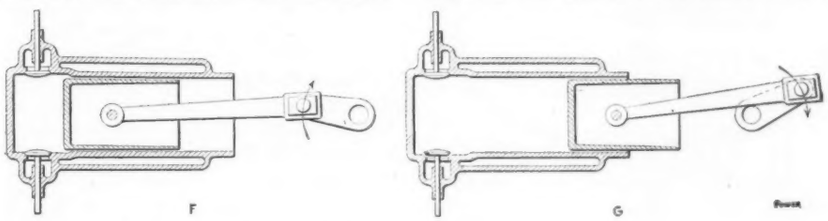


FIG. 4. THE THIRD STROKE; EXPANDING THE HOT GASES

In moving from the position represented at *D* to that at *E*, Fig. 3, the piston compresses the mixture in the cylinder, the valves remaining closed, as shown, and this is therefore termed the compression stroke. Just before the crank reaches the inner dead center at the end of the compression stroke, the igniter operates and the rise of pressure due to combustion begins; this continues while the crank is moving over the center, from the position shown at *E*, Fig. 3, to that shown at *F*, Fig. 4, the motion of the piston during this brief period being so small as to be negligible. This completes the third event (combustion) of the cycle and the fourth event, expansion, begins.

Expansion, like all the other events except combustion, has practically a full piston stroke of its own. It begins as soon as the crank passes the inner dead center (*F*, Fig. 4), and continues during the outward stroke until the exhaust valve begins to open, just before the end of the stroke, as indicated at *G*, Fig. 4. While the crank is passing from this position to that of *H*, Fig. 5, the burned gases escape to the atmosphere under their own pressure, which is some 15 to 30 pounds above that of the atmosphere when the exhaust valve first cracks open. By the time the crank reaches the outer dead center (*H*, Fig. 5), the exhaust valve is wide open, and as the piston moves back on the instroke again, the burned gases are pushed out of the cylinder by the piston, which is, of course, being driven then by the flywheel if the engine has only one cylinder. The exhaust valve remains open, as indicated at *I*, Fig. 5, until the end of the expulsion stroke, which is the movement of the piston from the position *H*, Fig. 5, to that of *J*, Fig. 6; then it closes and the inlet valve begins to open, ready to take in the next fresh charge and begin a new cycle.

Summarizing the foregoing detailed explanation, the four-stroke cycle consists of taking in a charge during one outward stroke, *A, B, C;* compressing the charge during one return stroke, *C, D, E;* combustion while the crank passes the inner center, *E* to *F;* expanding the heated gases to do work during a third stroke, *F, G, H,* and driving the burned gases out of the cylinder during a fourth stroke, *H, I, J.*

From this it should be clear that the name four-stroke cycle (commonly miscalled four-cycle) does not mean merely admission, compression, combustion, expansion and exhaust, but means that *these five events, constituting one cycle, are carried out in four strokes of the piston.*

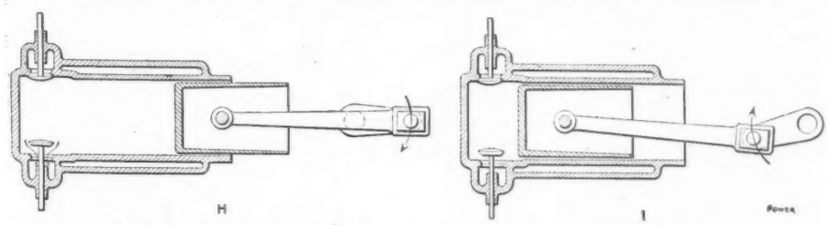


FIG. 5. THE FOURTH STROKE; EXPELLING THE DEAD GASES

THE TWO-STROKE CYCLE

It does not take much thought to make one realize that the five events of a cycle cannot be carried out completely in one cylinder during only two strokes of the piston. In the four-stroke engine one outstroke is devoted to suction and one to expansion, and both of these events cannot be performed during a single outstroke. Moreover, one instroke accomplishes compression and another pushes out the burned gases, and these two results cannot be produced in the same stroke.

In order to carry out the five events within two piston strokes, it is necessary to make three sacrifices: (1) The charge of gas and air must be pumped into the cylinder instead of being drawn in by the motion of the piston, because (2) the admission of each fresh charge must be accomplished within a very brief space of time between the end of the expansion stroke and the beginning of the compression stroke, and (3) exhausting the burned gases must be done within the same brief period of time.

The way in which these requirements are commonly met in actual engines is to open the exhaust port for a very few moments while the piston is at the end of the outstroke and at the same time force in the fresh charge under pressure supplied by an outside pump. The inrushing charge of gas and air (or sometimes air alone) helps to drive out the burned gases.

Figs. 7, 8 and 9 illustrate this method of carrying out the cycle. At *A*, Fig. 7, the piston is supposed to be just about to begin the expansion stroke, the compressed mixture of gas and air having been ignited just as the crank reached the dead center. The piston is forced by the expanding gases from the position at *A* to that shown at *B*, and when it reaches the latter position it begins to uncover the exhaust port and relieve the pressure in the cylinder. This practically completes the expansion stroke.

While the piston travels from the position shown at *B* to the end of the stroke, Fig. 8, and back again to the position *C*, Fig. 7, the burned gases are discharging through the port *E* and, at the same time, fresh mixture is entering through the inlet port *I*. The mixture is directed toward the cylinder head by a deflector *D* on the piston head, to prevent it from rushing across to the exhaust port *E* and passing out with the exhaust gases.

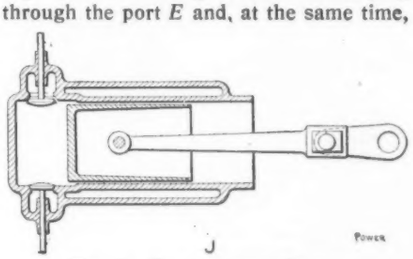


FIG. 6. END OF THE CYCLE

When the piston has covered the inlet and exhaust ports in its travel along the instroke (position *C*), compression begins, and during the movement of the piston from this position back to the position *D* at the end of the instroke, the charge is compressed more and more. When the crank reaches the inner dead center, with the piston in the position *D*, Fig. 7, the compressed charge is ignited

and burns while the crank is passing the inner dead center; then the events just described are repeated.

Summing up the foregoing description, the cycle is carried out as follows:

Combustion in the position shown at *A; expansion* while the piston travels from the position *A* to that at *B; exhaust* while the piston travels from the position *B* to the end of the stroke and back to *C; admission* during the same period with exhaust; *compression* during the instroke from the position shown at *C* to that at *D*. As these events all occur while the piston makes one outstroke and one instroke, the cycle is completed in two strokes and the engine therefore works on the two-stroke cycle.

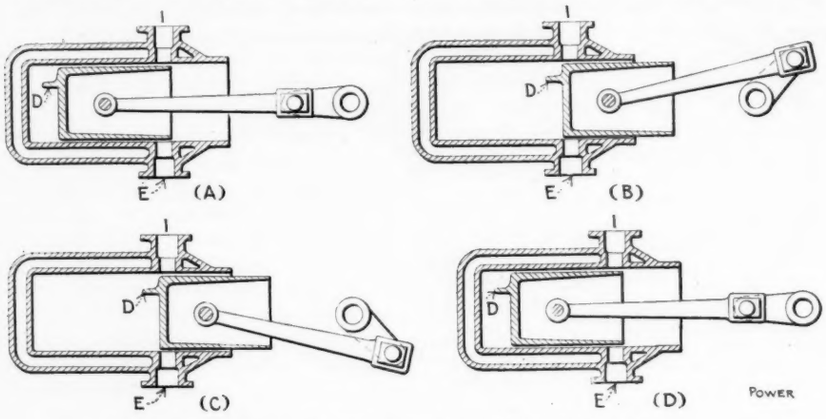


FIG. 7. DIAGRAMS ILLUSTRATING THE TWO-STROKE CYCLE

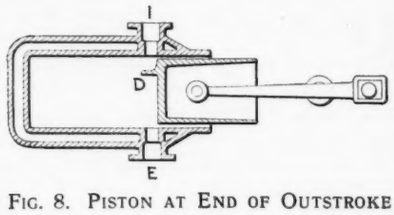


FIG. 8. PISTON AT END OF OUTSTROKE

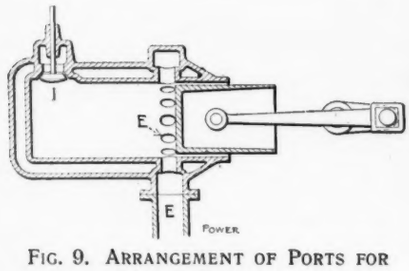


FIG. 9. ARRANGEMENT OF PORTS FOR LARGE ENGINES

From this description it should be clear that the difference between the four-stroke and the two-stroke cycles is that the suction and exhaust strokes of the four-stroke cycle are omitted in the two-stroke cycle, the discharge of burned gases and the admission of a fresh charge both occurring during the very short time while the piston is passing from the position *B*, Fig. 7, to the position of Fig. 8 and back again to *C*, Fig. 7.

Fig. 9 illustrates the principle which is commonly followed in the arrangement of ports used in large engines for controlling the admission and exhaust. Instead of covering and uncovering both the inlet and the exhaust ports with the piston, only the exhaust is controlled by the piston; the inlet port is provided with the same form of valve used in four-stroke engines. There are several exhaust ports, arranged in a belt around the cylinder barrel as indicated at E and when the piston has reached the end of the stroke, the inlet valve begins to open and admit a fresh charge. It remains open until the piston on its return stroke has covered the exhaust ports; then it closes and the further backward travel of the piston compresses the charge, exactly as described for the double-ported engine of Figs. 7 and 8.

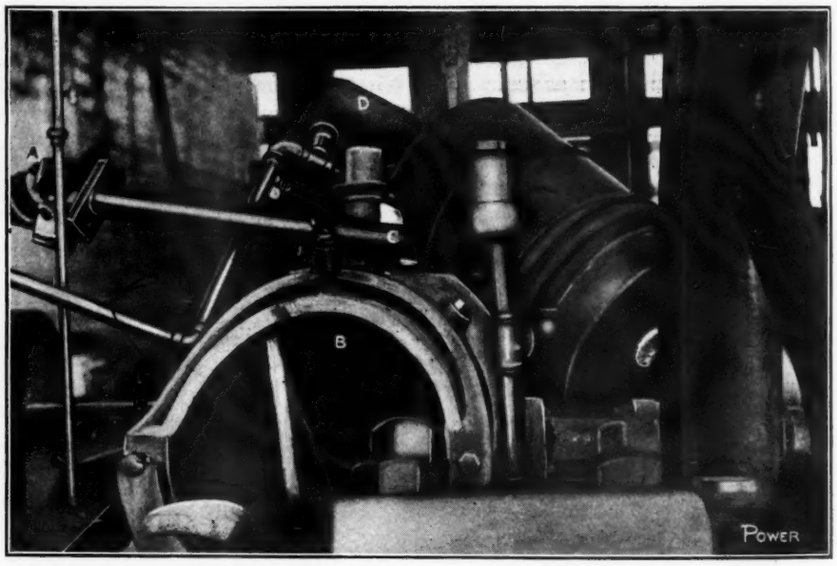
The only important differences in the operation of the two types are that the inlet port of the type in Fig. 9 is controlled by a valve instead of by the piston, and that therefore the charge is admitted at the opposite end of the cylinder from where the burned gases are discharged. There are many minor differences in the arrangement of details such as the control of the mixture proportions, the use of separate gas and air valves, etc., but so far as the essential features of carrying on the cycle are concerned, the two types operate alike. Both work on the two-stroke cycle and the timing of the various events is practically the same in both types.

A Bad Wreck from a Small Cause

BY M. W. UTZ

The accompanying photograph shows the wreck of a 14x20-inch single-cylinder horizontal gas engine of about 45 horsepower, which was used to drive a small machine shop.

The front brass on the crank pin had been cracked and a ¼x3-inch steel plate had been bolted over it for reinforcement. This plate and the brass broke, as shown at *A*, letting the connecting rod drop down, and when the crank came around the broken end caught against the crank, buckling the rod slightly and breaking a piece out of the cylinder 10 inches back and down to the center line of the cylinder, as shown at *B*. The connecting rod also broke a piece out of the piston 10 inches back and nearly half way across,



WRECK OF 45-HORSEPOWER GAS ENGINE

as is partially shown at *C*. The broken pieces of the cylinder and piston are shown lying in the cylinder, except the one piece out of the top of the cylinder, which is still connected to the overflow pipe of the jacket cooling water, and is shown at *D*.

The engine was running on natural gas, at 200 revolutions per minute, and delivering about 13 horsepower at the time of the accident. It had been in service between four and five years up to the time of the accident.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment

Practical Points on Electric Wiring

By Walter C. Edge

Nearly all engineers understand ordinary electric wiring, but there are several methods that may be new to the majority, or not thoroughly understood.

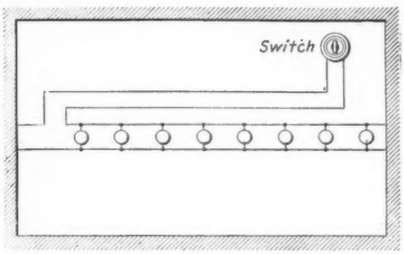


FIG. 1

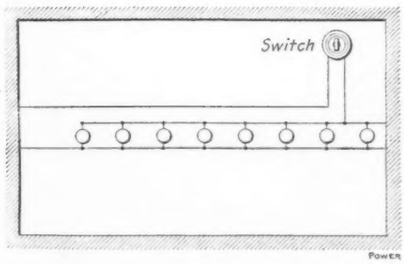


FIG. 2

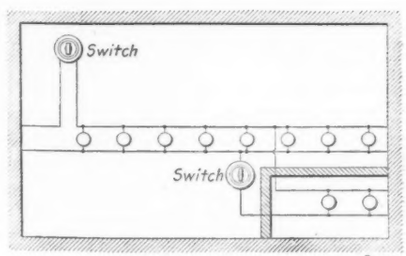


FIG. 3

Fig. 1 shows the usual method of connecting the control switch when the switch must be located at the opposite end of the building from the entrance. Fig. 2 shows a much better method, giving the same control with only three wires, instead of four, running the full length of the room.

In case there is a small room at the end of a large main room, for instance, a cold-storage room, three methods of wiring are available. Fig. 3 shows one that is sometimes employed. The fault with this lies in the fact that the main room must be lighted first in order to get light in the small room. Of course, key sockets would be used in this case, so that the lamps in the main room may be turned off individually if necessary. Fig. 4 shows a method used by most engineers for lighting the small room independently of the large one. Another circuit is carried from the entrance to the small room, making four wires instead

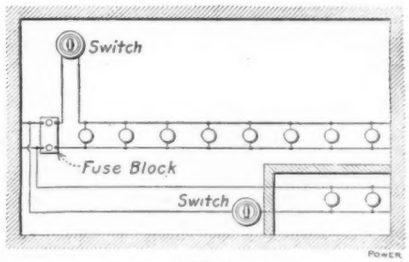


FIG. 4

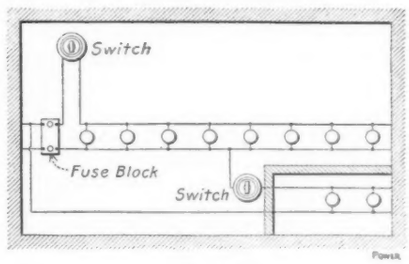


FIG. 5

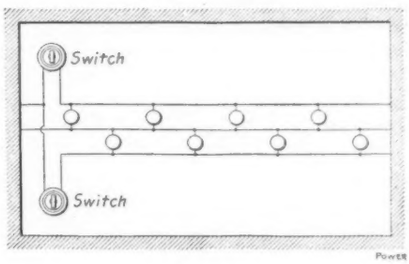


FIG. 6

of two extending the full length of the large room. The simplest and most economical way is to bypass the switch for the main room, as shown in Fig. 5. By wiring this way the lights may be turned on in the small room whether those in the main room are on or not, and it requires only three full-length wires instead of the four used in the previous method.

Occasionally it is necessary, for the sake of economy, to have the lights controlled by two switches instead of one, so that either one-half or all of the lamps may be turned on, as required.

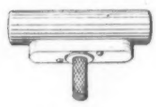


FIG. 7

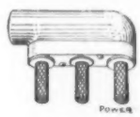


FIG. 8

FIG. 9

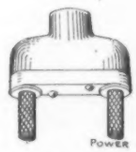


FIG. 10

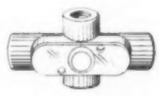


FIG. 11

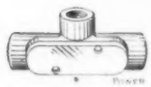


FIG. 12

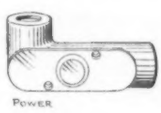


FIG. 13

FIG. 14

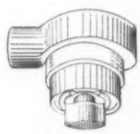


FIG. 15

FIG. 16

In a case like this the way to wire them is as shown in Fig. 6.

Conduit Work

Conduit work is largely replacing the open wiring, especially in large plants, owing to the danger of fire from the open wiring. There are so many fittings on the market now that this kind of work can be done very neatly and easily.

The fittings that are used the most are shown in Figs. 7 to 16 inclusive. Fig. 7 shows an outlet fitting for use in the service line and Fig. 8 shows the corresponding outlet fitting for use at the end of the service line. Fig. 9 shows another type of end fitting. Fig. 10 is an exceptionally neat fitting used generally on the end of a branch line running along the side of a beam or joist, neces-

sitating a bend at the end in order to clear the wood.

Fig. 11 is a cross used for branches to each side of the main. Fig. 12 shows a tee for a one-way branch from the main.

Fig. 13 illustrates a fitting called a "bendkick" used wherever a sharp bend is necessary. The back of the ell screws out, making it an easy matter to draw the wires through.

Fig. 14 is a fitting used on the end of an outside pipe run up the side of the building. Owing to the fact that it is practically impossible to enter wires in a fitting like this after it is in place on the end of the pipe, it is flanged and provided with a small corresponding

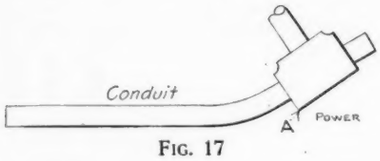


Fig. 17

flange which is screwed on the end of the pipe. The wires are drawn through the fitting separately and it is then bolted to the other half of the flange on the end of the pipe.

Figs. 15 and 16 are boxes to take snap switches; the snap switch is screwed to the supporter shown inside the box.

Upon starting on a job of conduit work the first thing necessary is a bending device. There are thoroughly satisfactory benders on the market, but if one of these is not obtainable, a fairly good bender may be easily made by screwing a tee on the end of a piece of pipe about 3 feet long. If ½- or ¾-inch conduit is used, the bender should be made of 1-inch tee and piece of pipe.

In making a bend, accuracy can be obtained by exercising a little care and

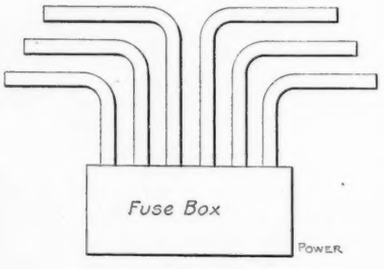


Fig. 18

judgment. First place the length of conduit on the floor, putting the end against a wall or something solid; mark off the required length and then slip the bender on the pipe to within about 2 inches of the marked point and bend the pipe about 20 degrees, as represented in Fig. 17; let the bender slip back a few inches and bend a little more, repeating this until the proper bend is obtained.

All bends should be made with as large a radius as possible, so that the wires may be drawn through without trouble. If an offset is to be made in the conduit it should be made with as gradual a bend as possible.

When several circuits lead from a fuse box, it is best to take as many

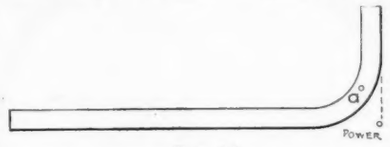


Fig. 19

lengths of conduit as there are circuits and start on all the circuits at the same time, rather than run them separately, as the bends can be made alike with greater ease than is possible if put up separately. The pipes must be bent at different distances from the fuse box in order that they may all lie in the same plane. If half of them turn one way and half the other, there should be two pipes bent to each length, one in each group, as represented in Fig. 18. If the pipes are to be 6 inches apart, for example, the second pipe must be bent at a point 6 inches beyond the bend of the first one,

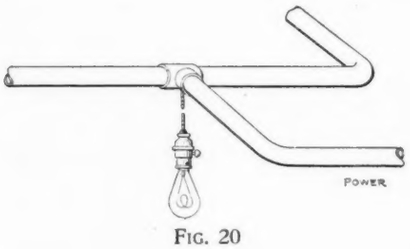


Fig. 20

the third one at a point 6 inches beyond the second one, and so on, as shown in Fig. 18.

It is better when bending conduit to put the mark on the floor than to mark the pipe; then the pipe can be bent accurately to the mark, as indicated in Fig. 19, whereas the mark would be about at *a* if placed on the pipe.

Considerable money can be saved by using a little forethought in this kind of work. For instance, if a light is to be placed near a branch, the tee can be used for the light as well as the branch, as indicated in Fig. 20, saving a condulet fitting at the expense of a few feet of conduit.

After the mechanical work is complete a steel-fishing wire is pushed through the conduit, the conductors are fastened to the end of it and are drawn through the conduit. A little powdered soapstone should be rubbed on the wires or blown through the conduit to make the wires slip through more easily.

The conduit should be grounded; water pipe makes the best ground, but if no water pipe is close enough, a pipe can be driven into the ground.

Kootenai Falls, Mont., to Be Developed

Joseph A. Coram, owner of the water-power site at the Kootenai falls, twelve miles west of Libby, Mont., has announced his intention to at once proceed with the organization of a company to develop the immense water power there, the object being to supply power not only for Lincoln county industries, but for some outside points also. The details have not been entirely worked out.

The water in the Kootenai river, which is the largest stream in Montana, according to Government reports, goes through the Purcell mountain gorge in Lincoln county, Mont., in a succession of falls and rapids. The horsepower that can be developed is estimated at 200,000 at low-water mark, and goes to as high as 357,000 during high water. There are no great engineering difficulties to overcome in connection with the work.

Hydroelectric Development of Deerfield River

Interests connected with a Connecticut river power company have completed plans for the hydroelectric development of the Deerfield river, and have issued $5,000,000 bonds to carry on the work.

The first stages of development will provide for the generation of 25,000 horsepower. It is intended to dam the Deerfield river in four places near Shelburne Falls, Vt., and build a reservoir at Somerset, covering an area of about six square miles. The water will be impounded to a hight of 100 feet.

It is said that this undertaking will ultimately cost $12,000,000.

Conservation of Water Power

Investigations of possible sites for developing water power on the public domain are being pushed by the United States Geological Survey, with resulting withdrawals of land from entry where it is found that valuable sites exist.

In July of this year 31,725 acres of such land were withdrawn, including a great number of power sites. No estimate has been made of the horsepower involved, but owing to the character of the power sites withdrawn it is believed to be very large. These July withdrawals make a total outstanding area withdrawn of 1,546,258 acres, based on the examination and recommendation of the Geological Survey, and involving thousands of power sites and doubtless millions of horsepower.

The withdrawals are made in aid of proposed legislation by Congress which shall provide for the fullest possible development of these enormously valuable properties and at the same time guard the public interests.

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words wanted

Cylinder Oil for Hot Bearings

My purpose is to show the advantage of having a clear idea of what to do in cases of a hot bearing, so that precious moments may not be lost.

I gave a little talk to my oiler and had a can of cylinder oil set apart and called it the fire extinguisher. The possibility of having a hot bearing seemed remote, but I explained to the oiler that cylinder oil would stand a considerably higher temperature than engine oil and that if cylinder oil were applied promptly and in sufficient quantities the babbitt might be prevented from running and the smoke stopped. If the trouble is discovered in time the engine may be continued in operation.

The opportunity to try the outfit came sooner than I expected. I was working with my back toward the engine and the oiler was busy wiping a machine when the oil-supply line got air-bound. The sense of smell warned me that there was a hot bearing around. I called for the "fire extinguisher" and had the reserve oil cups running before the fire brigade arrived, but it was the splash of the half pints or so of the heavy cylinder oil that absorbed the heat and penetrated underneath the brasses and stopped the smoking. Another engine was started up, and on turning the distressed engine over slowly I found that the brasses would not take oil freely. The oil grooves were cleaned out and the brasses scraped a little.

A. Z. McLeod.

New York City.

Turbine Oiling Trouble

I recently visited a steam plant just in time to learn a new stunt. Several large turbines were in operation, one of which was giving trouble. The oil on both ends of the bearing on the steam end was running out in a stream. The turbine was shut down and the operator was trying to locate the trouble. He asked me what I thought of it, and I suggested taking off the manhole cover on the exhaust end of the turbine. This disclosed the cause of the trouble.

The two 1½-inch pipes used for a seal supply had about 6 inches of lime and scale around them. Both pipes were taken out, and the pipe which carried the water from the bearing was found to be plugged with scale, thus causing the water to flood the bearing and mix with the oil, and run out of both ends of the bearing.

The pipe was cleaned out and put back again, but the operator's troubles were not over, as the water had gotten down into the oil tank of the turbine, and before the machine could be put into operation several barrels of new oil were put in the tank.

T. E. Jones.

South Bend, Ind.

Condemns License Law

It is to be hoped that in their efforts to obtain license laws in the various States, the engineers' associations will show more sense and less bigotry than the engineers in Philadelphia did when they had a license law passed in that city. The law in that city deprives men of their right to earn a living and consequently of their liberty as American citizens, if they have no influential friends or will not ask favors.

Before an engineer is granted an examination for a license to operate a steam plant in Philadelphia, he must obtain vouchers from two licensed engineers of Philadelphia; engineers of any other part of the United States are not good enough. Two vouchers from employers are also necessary, and a fifth voucher must be signed by two other persons, testifying to the good character of the applicant.

This would be all right for an apprentice boy or a man who has never passed an examination and has neither a marine nor a stationary engineers' certificate, but if an engineer can produce good recommendations as to his character and ability and holds a certificate as chief engineer of ocean steamships or a first-class engineers' certificate issued by the civil-service board in one of our large cities, which entitles him to take charge of any power plant in that city, he should at least be entitled to an examination without being compelled to ask favors.

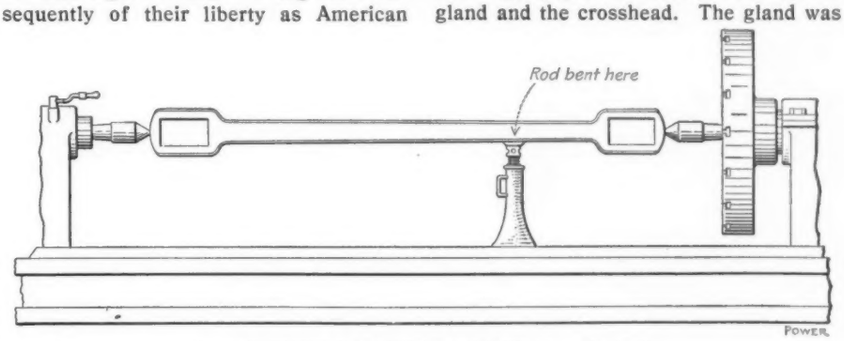
Of course, the engineers' associations in this "City of Brotherly Love" have been instrumental in having this absurd law passed in order to debar others from obtaining licenses, hoping thereby to reduce the supply of engineers and increase their wages. How well they have succeeded is shown by the fact that many of them work 12 hours for the same wages paid to coal passers on American ships for eight hours' work.

H. J. Leiper.

Philadelphia, Penn.

Straightening a Connecting Rod

An engineer, while tightening the stuffing-box gland of a 250-horsepower Corliss engine, got his wrench between the gland and the crosshead. The gland was broken and the connecting rod was bent.



Straightening Connecting Rod in a Lathe

The rod was placed between the centers of a heavy engine lathe, as shown in the accompanying illustration. The rod was heated to a cherry red, and a slight pressure with a jack was sufficient to bend it. After several attempts the rod was straightened, then allowed to slowly cool.

Before the rod was put back on the engine, it was polished where it had been heated and was apparently in as good condition as before the accident.

C. A. Gilson.

East Lansing, Mich.

Belt Ran to Side of Pulley

Some time ago I helped to set up and start a new horizontal waterwheel, which had a 60-inch pulley on the shaft. On starting up the belt ran off 2½ inches to one side of the pulley, which had a 20-inch face, although the wheel shaft was in line with the main shaft in the mill.

After indulging in much thought and worry, I took a steel tape and measured the circumference of the wheel pulley each side of the face and found it to be about 3/16 inch larger on one side of the face than on the other. The pulley was turned half-way around and put back on the shaft, the belt replaced and when the pulley was started the belt took the center as nicely as one could wish. This, of course, was a crown-face pulley and in turning up the face care had not been taken to turn both sides of the center to the same diameter.

Later, I helped to set up and start a 400-horsepower cross-compound engine with a 32-inch-face flywheel. On starting the engine the 30-inch belt ran off one side of the flywheel about 3 inches. There was some guessing as to what caused it.

I found the flywheel to be the same size on both sides of its face, but the 4-foot pulley on the jack shaft was 1/16 inch larger in diameter on one side of the face than on the other. The pulley was turned half-way around and put back on the shaft. The next morning the belt took the center of the flywheel and the driven pulley, much to the surprise of the other workmen.

JAMES MITCHELL.

North Adams, Mass.

Improved Stop Valve

Almost every valve used in pipe connection in an engine room leaks more or less. I have had troubles with valves in boiler rooms which I found were caused by faulty construction of the valve and not poor management on the part of the engineer.

In the ordinary stop valves I have used the bearing surface between the valve and its seat is one conical surface and if the valve wears irregularly or a bit of dirt gets in the bearing surface, the valve will soon begin to leak.

To remedy this defect I designed the valve shown in the accompanying illustration, which is a sectional view.

The valve disk is made with an annular inverted V-shaped groove which fits a corresponding elevated surface on the valve seat.

This construction gives the valve disk and seat two bearing surfaces, so there is little chance for steam leakage. If the valve leaks at the inner surface, the steam expands in the cavity of the groove and forms a water packing, and thus prevents further leakage.

The distance of the center of the groove from the center of the valve differs slightly from that of the center of the valve seat's head. When the valve disk is pressed against the seat on the outer edge it springs in to some extent and gives a uniform pressure on the valve seat, thus keeping the valve nonleakable.

YAEKICHI SEKIGUCHI.

Tokio, Japan.

Daily Log Sheet

The accompanying daily log sheet is a duplicate of those used by the Mobile Light and Railroad Company, Mobile, Ala.

The plant of this company is said to be one of the most efficient in the South, and its system of keeping records of all operating expenses is the most complete I have seen.

S. KIRLIN.

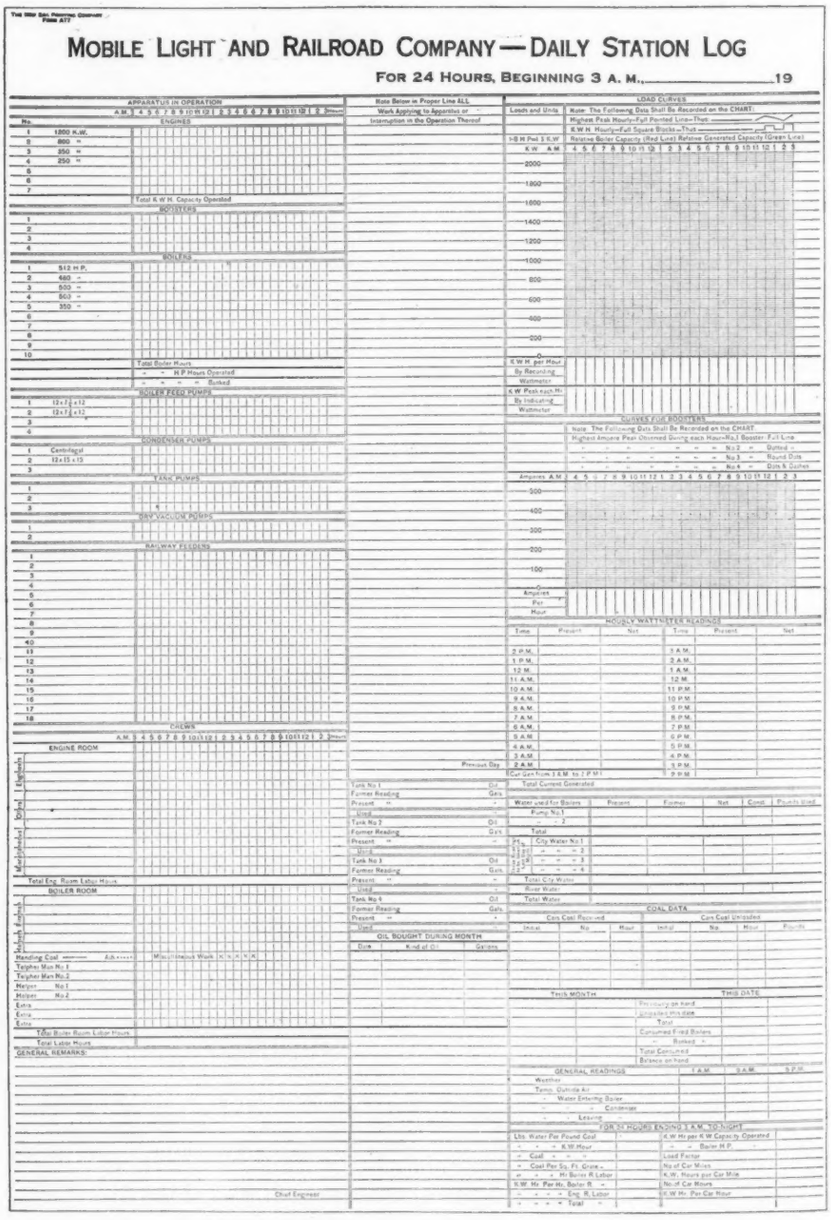
Mobile, Ala.

No Relief Valve on Heater

I took charge of a plant some time ago in which a closed heater was used, but there was no relief valve on the heater or feed line.

I ordered a relief valve as soon as I had looked over the piping, but the owner refused to get it. He said that the plant had run five years without one and he did not see why he should get one for me.

A few days later the owner sent me out of the city to look at a second-hand engine he thought of buying, the superintendent running the plant in my absence. After I arrived at my destination I re-


MOBILE LIGHT AND RAILROAD COMPANY—DAILY STATION LOG

FOR 24 HOURS, BEGINNING 3 A. M., 19

DAILY LOG SHEET FOR POWER STATION

ceived a telegram to "drop everything and hurry back; break down."

Returning, I found the steamfitters taking down the heater and part of the feed line. The superintendent had filled the boilers to three gages of water before shutting down for noon. He shut the feed valve on the boiler and stopped the feed pump, which pumped direct from the city water; there was a check valve on the discharge pipe next to the pump. The feed line had filled up with cold water while the engine was shut down for dinner. As there was plenty of water in the boiler at one o'clock, he started the engines without opening the feed to the boilers. The engines had only run a short time when the exhaust going through the heater caused the water in the feed line and the heater coils to expand and burst both the heater coil and the feed pipe.

I explained things to the boss and showed him where a few dollars spent for a relief valve would have saved him the $250 that the shutdown cost him.

I now have a relief valve on the feed line and one on the feed pump and a spare one in stock.

W. V. FORD.

Norwich, Conn.

Isolated Plant Practice

This letter deals with the cost of producing power in an isolated plant where I was in charge some time ago, and is submitted in the hope that it will strengthen the cause of the isolated-plant engineer. The wattmeter diagrams speak for themselves and show a good performance for a 300-kilowatt plant.

Figuring the coal at $2.90 per ton of 2240 pounds, the coal consumption for the day, the chart shown in Fig. 1 was taken, cost about $31, that for Fig. 2, about $34.80 and for Fig. 3, about $35. The rate of wages were, chief engineer, $3.14 per day; two assistant engineers, $5.16; two firemen, and one general utility man $6. The removal of ashes cost $1 per day; oil $0.25; waste, $0.10 and water $1. Allowing 10 per cent. for maintenance of equipment and 5 per cent. on a capital of $30,000, which is the cost of the plant, the following is obtained: Maintenance of equipment, about $8.50, and interest $4.25 per day, or about $13 in round numbers, which gives a total of $60.55 per day. Now, according to my figuring, the cost per kilowatt hour for Fig. 1, is slightly over 1.6 cents; for Fig. 2 and for Fig. 3 it was slightly over 1.3 cents.

One day about three years ago two central-station representatives came into the engine room and as they had permission to look over our records for the previous years, I gave them access to the records dating from July 1, 1907 to July 1, 1908. They spent two days looking them over, but I have not seen them since. Shortly afterward, the superintendent told me they offered to furnish current at 4 cents per kilowatt-hour.

The coal consumption per kilowatt-hour is heavy; about 6 pounds. With more efficient boilers, I could have made a better showing.

A. C. KIERMEIER.

Philadelphia, Penn.

A Pump Experience

The action of a 400-horsepower triple-expansion engine driving a hydraulic dredge puzzled me for a time. It was fitted with a throttling governor and under normal load turned over about 200 times per minute. The engine was directly connected to a centrifugal pump which had a 22-inch discharge and an open-end impeller. The discharge line varied from 200 to 1500 feet or more. The head would never be over 18 feet, but it would vary with the tide.

Occasionally the pipe line would break or pull apart and the engines would slow down; why, I could never understand. It would also labor harder on a short discharge than on a long one. With a heavy load the pressure would go up but little with a considerable increase in speed.

As the line became longer the load became heavier. With a heavy load the water would slip back over the side of the vanes and churn, reducing the wheel resistance and causing it to act much the same as the drivers on a locomotive when slipping under a heavy load. When the load was reduced this slipping at once ceased and the vanes forced the full volume and made the engine pull harder.

Another pump was finally installed with the inclosed type of runner, which is similar to those used in multiple-stage pumps. Although this did not entirely eliminate the slip, a marked improvement was noted.

THOMAS H. HEATH.

Seattle, Wash.

Questions for Discussion

I wish to ask a few questions and trust all replies will be based on practical experience. They are as follows:

Will turning cold water into a red-hot boiler cause an explosion, and if so, why?

Will cutting two or more boilers in together without having the pressure equal on each boiler be liable to cause an explosion provided that ordinary care is used and the valve is slowly opened; if this practice is dangerous, why?

If one boiler of a battery of several explodes, is it liable to cause the other boilers to explode; if so, why?

Suppose a condensing engine should be suddenly relieved of its load and attain a dangerous speed. If the throttle valve is closed tight but the condenser continues to operate and maintain the usual vacuum, will the engine increase in speed after the throttle is closed; if so, why?

Alton, Ill. H. R. ROCKWELL.

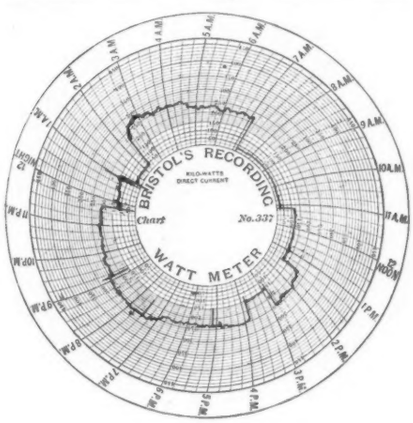


FIG. 1

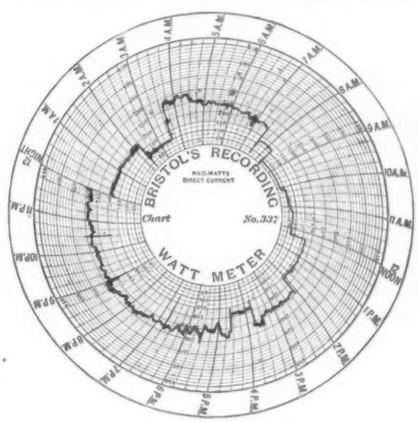


FIG. 2

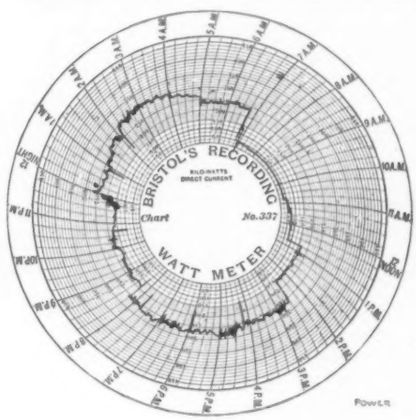


FIG. 3

THREE ISOLATED-PLANT RECORDS

Prevent Standpipe Freezing

I would like to ask some of the readers of POWER, what is the best remedy for preventing a standpipe from freezing? It runs from a 20,000-gallon iron tank, elevated about 50 feet above the ground and is situated in a very exposed place. If a covering will answer, what kind is best to use?

THOMAS NICHOLSON.

Newark, N. J.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Direction of Compressor Rotation

On page 189 of the August 1 issue, D. C. R. asks if there is any particular reason why an air compressor should run under. The answer given is, "There is none. In fact, the friction will be slightly less when running over."

It seems to me that this is not quite accurate for a steam-driven air compressor, especially if the cutoff in the steam cylinder occurs before half stroke. Under these conditions the flywheel is doing the greater part of the work; therefore, it is the same as a belt-driven machine, and when running under the greatest stress comes on the bottom guide. I should think that this would produce less friction than when the top guide takes the thrust.

L. C. TUCKER.

Newburyport, Mass.

[The question and answer relate to the compressor alone. The effect of the angularity of the connecting rod when running over is a tendency to lift the crosshead from the lower guide on both strokes, which will reduce the friction slightly, the guide being partially relieved of the weight of the crosshead. The friction due to the thrust of the piston rod or to the pull of the connecting rod is exactly the same, no matter which guide takes it.—EDITOR.]

Filling Oil Storage Tank

In the August 15 issue, W. W. Warner asks for information as to the best method of emptying tank cars. All oil-tank cars have an outlet at the bottom which will take about a 3-inch hose coupling. One method is to run the oil by gravity from this bottom connection, with the vent in the dome open, as the storage tanks were below the level of the car.

In Mr. Warner's case the tanks are above the car so that he can connect his discharge to the bottom outlet and run a ¾-inch air pipe to the pipe connection on the tank dome and force the oil out by air pressure. I have assumed that he has compressed air as he speaks of forging furnaces.

Oil-tank cars (some of them, at least) are equipped with heating coils so that when the oil is very stiff in cold weather steam can be turned on to heat the oil. This makes it as easy to handle in winter as in summer. Steam can be used to force the oil out, but the condensation will go into the storage tank and will have to be drained off after it settles to the bottom.

I believe tank cars are tested to withstand 40 pounds pressure, but I would not use over 10 pounds if this pressure will do the business.

I advise him to run the steam, air and oil pipes up to a convenient place and use a short section of hose to connect to the car as it may not always be left in exactly the same spot.

Mr. Warner states that the tanks will be 5 feet above the ground and 3 feet above the car. If he could bring the tanks down on the ground, below the level of the car, he could transfer the oil by gravity. This would be the easiest and the cheapest way.

Fuel oil is not very thick, even in freezing weather, and is quite easily pumped.

I have transferred sulphuric acid from a tank car to a storage tank at a higher level by air pressure. Acid tanks have two pipe connections in the dome; one leads to within 1 inch of the bottom of the tank, the other over the acid. The discharge pipe is connected to the pipe leading to the bottom of the acid and air pressure is used to force the acid out. Care must be taken to keep water out of the acid, and an air pressure of over 25 pounds must not be used, as either may cause an explosion.

There is but little danger from oil, but I would advise as low pressure as will do the work.

J. C. HAWKINS.

Hyattsville, Md.

Replying to W. W. Warner's question in the August 15 issue, oil may be transferred from tank cars to storage tanks by connecting a pipe between the supply tank and the storage tank and introducing an air pressure in the supply tank. This is a cheap and efficient method if compressed air is available. If not, I suggest the following for Mr. Warner's supply tanks:

If they are outdoors, bury them in the ground below the freezing depth. Place a manhole over the tanks so that they may be accessible. A telltale should be placed in them, and also a heating coil in the event of low temperature. Oil may be taken from the tank by a rotary oil pump and delivered to the forging furnaces. This type of installation should give entire satisfaction.

R. G. Cox.

Milwaukee, Wis.

In reference to W. W. Warner's letter in POWER for August 15, I would say that a quick and cheap way to get the oil into his storage tanks is to connect a hose or pipe from the discharge of his pump to the bottom of the tank. Then make a similar connection on top of the tank to the storage tanks. Be sure that the top connection is opened before the pump is started; if there is no safety valve on the tank an overpressure on the tank will cause disastrous results. The water will force the oil into the storage tanks and the oil will always float on top of the water. I have tried this experiment several times and it worked successfully.

PATRICK MALLOY.

New York City.

Silent Running Engines

The editorial in the August 8 issue under the title, "Silent Running Engines," is the most comforting piece of reading I have seen for many a day. Generally, when the subject of compression is touched upon it is usually touched upon with a 20-pound sledge; that is, you must either have too much or none at all. Heretofore there has been no half-way point at which we were allowed to run our engines, the advocates of compression holding that if a little is good, a lot of it is better, so give enough to bring the blood out of the keyways in the cranks and flywheels. On the other hand, the advocates of the noncompression theory contend that if too much compression will loosen the keys and spokes of the flywheel, one should have none at all; in fact, if it is possible to get a 30-inch vacumm in the cylinder at the end of the stroke, do so. Now comes this editorial giving all of us a grain of comfort by telling us to have just enough. So be it.

Compression, of course, can be figured out very nicely by taking into account the

momentum of the moving parts just before they must be brought to rest, but I venture to say that the quietest-running engines in the country are those in which the right amount of compression is ascertained in the same way as the Arkansas traveler plays the fiddle—by ear. As has been stated, more noise is caused by faulty admission in a good many cases than by faulty compression. Once I stepped into an engine room where there was a 2000-horsepower cross-compound engine which was quite noisy on the high-pressure side, although one could still hear the dashpot drop if he were close enough. I was standing at the rear of the high-pressure cylinder when an extra load came on the engine. I could hear the piston tap the cylinder head at each revolution, so I stepped to one side and started to go away. It had never hitherto occurred to me why an engine should pound more on a heavy load than on a light load and I never ran across a man who seemed to know. Many seemed to be willing to risk a guess, but the guess was usually that if the valve opened wider on a heavy load than on a light load the steam pressure was greater on the piston at the point of admission. This, of course, is all wrong, as it is only possible to get boiler pressure on a Corliss engine, and you get that regardless of the load.

The lead on some engines certainly has a good deal to do with the amount of noise they will make. It is not always possible to have an engine run quietly at all loads; all that can be done is to get the best running at the load the engine is to carry and it will not cut up very badly at a slight over or under load. One may indicate a half day if he will and get about what he wants in the way of a diagram and still have a disagreeable pound. Simply loosen up the back nuts on the link rod and dashpot rods, start up, and put the normal load on the engine. Then try a little less or a little more until you think you have what you want, and then shut down and tighten up the lock nuts again.

This may not be scientific, but it does the business and if it affects the economy I have not yet noticed it. It is found on a 46 and 94 by 60-inch engine running at 75 revolutions per minute that about 1/16-inch lead on the high-pressure side is all right, and on the low-pressure side the steam valves set at line and line, or with no lead, give the quietest effect. The exhaust valves close at 5 inches from the end of the stroke. This engine makes a kilowatt-hour on 17 pounds of steam, so I do not see any objection on that score. The compression on the low-pressure side with this setting runs up to about 10 pounds absolute. It is very evident that if with a 25-inch vacuum you could get a final compression on the low-pressure side of 10 pounds absolute, if the engine should begin to exhaust to the atmosphere the compression would run up in proportion to the ratio between atmospheric pressure and condenser pressure, which in this case means 60 pounds gage. It would seem, then, that if the receiver pressure was not raised, the exhaust valves would necessarily become a sort of relief valve to let the excess compression back into the receiver. It does not take much slamming due to this condition to break a valve, as one weighs 1800 pounds. To avoid trouble a rig was put on the low-pressure governor gear so that the receiver pressure could be immediately raised enough to hold the valves down on the seats.

E. H. LANE.

Kansas City, Mo.

Steam in Cold Boiler

I have read, in the August 15 number of POWER, Leon Roundy's account of how a boiler was tested for leaks by filling it with cold water and then turning live steam at 100 pounds pressure in on top of the water. The chief in a plant where I was working filled a vertical water-tube boiler with cold water and started the fire. When this was done he cut the boiler in. Nothing happened, but I would have preferred that this work be done when I had a more distant view. What did the chief think was gained by doing this? This boiler certainly could not make any steam until the water reached a temperature which would correspond with its pressure and in cutting it in cold it only added a load to the other boilers for a time, to say nothing of the evils of unequal expansion.

In Mr. Roundy's case, assume that the water filled the lower half of the steam drum and that the cold water had a temperature of 60 degrees. Then the plates below the water would have the temperature of the water, or 60 degrees. The plates above the water were in contact with steam of 100 pounds pressure with a temperature of about 338 degrees. Thus, the two parts of the drum had a difference in temperature of

$$338 - 60 = 278 \text{ } degrees$$

Sames, in his "Mechanical Engineers' Handbook," gives the coefficient of linear expansion of mild steel as 0.0000057 and the coefficient of elasticity as 30,000,000. Using these figures, each inch of the plate above the water would expand

$$0.0000057 \times 278 = 0.0015846$$

of its length more than the same length under the water. When expanding, the plates will exert as much force against anything which tends to hold them back as would be required to stretch them the same amount if the temperature remained constant.

If we use the formula for the amount of elongation of a piece under stress and transpose it to find the stress applied, we may find the stress necessary to stretch the plate the amount of expansion. If we let

P = Stress;
L = Length, in this case 1 inch;
A = Area, in this case 1 square inch;
E = Coefficient of elasticity;
X = Amount which the plate expanded; we have

$$P = \frac{X\,A\,E}{L} = \frac{0.0015846 \times 1 \times 30{,}000{,}000}{1} =$$

47,538 *pounds per square inch*

of section. This is the force with which the plates above the water expand. This strain must be taken up in the plate along the water line.

Of course, I do not mean to say that each square inch of the plate has this strain of 47,538 pounds, for the plate will not have a cross-section of 1 square inch for each inch in width. If the plate was ½ inch thick the cross-section would be ½ square inch and the strain would be 23,769 pounds. But the tensile strength of the same amount of plate would only be

$$55{,}000 \div 2 = 27{,}500 \text{ } pounds$$

The ratio of the strain due to unequal expansion to the tensile strength would remain the same.

If Mr. Roundy's chief filled the boiler with water under pressure, he might have filled it a little fuller until the air which had been trapped in the boiler was compressed to the required pressure. This would have shown what he wished to know. The fact that the other boiler was running leads me to believe that the water under pressure could be had.

L. A. FITTS.

West Fitchburg, Mass.

Central Station versus Isolated Plant

In reply to Mr. Schneider in the August 8 issue, I offer the following:

At my plant the boiler is but eight years old, and shows no signs of deterioration, and the character of the feed water is good. The flues are blown with a good blower every day and scraped once a week. Two gages of water are blown down daily and once a month the boiler is blown off and thoroughly washed out. It is impossible to keep a boiler any cleaner than that one is kept. The draft is ample, the setting is in good shape, and ashes and combustion-chamber soot are not allowed to accumulate.

There are no facilities for checking up. The only thing I could do was to clean off the firing floor and take a weighed load from the team, note the time and date firing on that lot began and when the last was used; then divide it off into an average number of barrowloads used in the day to that at night. While this is not very accurate, it is as close as it is possible to figure. My somewhat more

than two tons amounts to 4200 or 4300 pounds in 10 hours.

I assume that Mr. Schneider's boiler and engine are close together. They must be for him to be able to fire as low as 3½ pounds of coal per indicated horsepower-hour. However, considering the circumstances mentioned in my previous article, I am not dissatisfied with my showing. As for heating with exhaust steam, nearly every plant now uses that system (the one under discussion included) and the really uptodate ones use a vacuum pump in addition.

EMMET BALDWIN.

Sturbridge, Mass.

Lifting Water in Boilers

I noticed a letter by C. J. Harden, in the August 8 issue of POWER on "Lifting Water in Boilers." I wish to take exception to his statement that "A boiler on the line has its water in circulation and only a part of the water is at or very near the right temperature to flash into steam."

It is the generally accepted theory among engineers that practically all of the water in the boiler is at the temperature corresponding to the pressure of the steam. This must be so when we consider that above the water there is a large volume of steam which would immediately condense and give up its latent heat and heat up any water that happened to be near it which was at a temperature lower than the steam itself. Furthermore, the volume of water fed into a boiler is so small as compared to the total volume contained in the boiler that the injection of this comparatively low-temperature feed water has practically no effect in lowering the temperature of the water in the boiler. That is, it immediately mixes with the high-temperature water in the boiler and attains a temperature corresponding to the pressure.

Again, I note that Mr. Harden claims "pressure and temperature rise quickly and the boiler shell, not having time to adjust itself to the new condition, lets go." It seems highly improbable that this could occur, inasmuch as the boiler under discussion is directly connected to the steam main, which, according to the assumption in the beginning of the article, is at a pressure lower than that in the boiler; therefore, the boiler should certainly equalize the pressure through the steam pipe.

The point that I wish to bring out is the desirability of using automatic valves on boilers. These valves are now so built that they serve the double purpose of being automatic cutoff valves in case of a break in the boiler or steam line, and also of automatically cutting in a boiler when steam pressure is being raised. That is, when a boiler has been cut out of service and the pressure in it is lower than in the steam main, this valve will automatically close and isolate that boiler. Then as soon as the steam pressure is again raised in the idle boiler the valve will remain closed until the pressure in the boiler being cut in is just equal to that in the steam main or a pound or two higher. As soon as the latter pressure is reached the valve opens and the new boiler is thrown into service automatically, thus affording safety to both the steam main and the new boiler, and to all the other boilers as well.

It seems strange that these valves have not been more generally adopted in this country. In France automatic cutoff valves have been compulsory since 1886, when the explosions of Eurville and Marnaval liberated steam which killed 30 people and seriously injured 50 others. In other European countries such laws are now under consideration with a prospect of being passed at an early date. These automatic valves are recommended by the leading boiler-insurance companies of this country.

F. J. MCMAHON.

New York City.

Trouble with Leaking Tubes

In reply to Mr. Reimers' letter under the above heading in the July 18 issue, I have this to say: I have charge of a two-sheet, double-riveted lap-seam return-tubular boiler that has been in use 13 years. It has never developed a leaking tube yet, and the tubes are in perfect condition. This boiler has no rods through it below the tube line, as described by Mr. Reimers.

Why is it that the boiler inspectors tell us that a two-sheet boiler is a poor thing?

C. J. WRIGHT.

Alliance, O.

Riverton Turbine Accident

The account of the Riverton turbine accident in the issue of August 8 is interesting.

The theory advanced that the machine was running above normal speed seems the most tenable, and will, no doubt, stand until tangible proof is brought forward to the contrary.

At half speed or even full speed, with such rugged construction and generous factor of safety, there is hardly a possibility of a machine going to pieces, in the manner described.

It is inconceivable that a chisel or nut could have caused the disaster, as trouble from either one would have been observed at the moment of starting. Furthermore, as the stock of a chisel is ¾ inch or more and the assembling nuts are either 1¼- or 1½-inch bolt size, the small space between the bottom of the diaphragm and the top of bucket wheel would not permit the insertion of a chisel, much less a nut. Foreign substances in the intermediate holders would be out of the question also, as the parts could not have been assembled at all.

Again, any obstruction in the turbine sufficient to cause the accident would have made itself known on starting by a fluctuation in the step pressure, by vibration and by excessive first-stage pressure.

If the bucket wheels were rubbing, due to too high or too low a step screw, it would, if run long enough, wreck the internal parts of the machine, but would not cause such an accident as that at Riverton. If the machine were running at half or even at full speed, it would slow down first.

If tools or machine parts had been left in the turbine they would have been missed, so the obstruction theory might as well be dismissed.

It is not at all likely that a machine would run for 10 minutes, as stated, with the throttle cracked and remain at the same speed; it would most likely increase in speed. Judging by the meager particulars at hand, we would not be far out of the way in accepting the theory that the turbine was above its normal speed; that the governor failed to close the valves or that the valves were improperly set; that the emergency governor failed to work or, having worked, the throttle failed to close. The latter is quite possible in certain types of throttle valve if they are in other than their wide or nearly wide-open position.

That the buckets show signs of having rubbed does not necessarily mean that they were obstructed or that they were not central on starting up. A hot bearing or tight carbon packing would cause rubbing, or, more likely, a high speed would bring the greatest pull on the bucket segments along the line of the center of gravity, starting the rivets and causing the segments to turn upward or downward as the case might be.

It is not my intention to pass judgment but rather to share my limited experience with others.

C. A. BLUE.

Boston, Mass.

Locating Keyway in Corliss Valve Stems

S. Kirlin, writing under the above heading in the August 1 issue, explains his way of determining the proper position for the keyway in new valve stems.

I think it is better to lay the old stems on a good faceplate and locate the position of the keyway. Then it is an easy matter to mark the new stems. If the old stems are incorrect the proper position should be located and marked before the stems are removed. Then the procedure is as before, except that the gage will be set on the old stem to the proper position rather than to the actual position of the keyway.

I was interested in his tables, showing various sizes of lap for steam valves.

They may do when starting, but most likely the indicator will point out many defects that can be overcome that will change the lap, and the opening of the exhaust.

L. JOHNSON.

Exeter, N. H.

Centrifugal Pumps, Their Design and Construction

In the August 1 issue of POWER, on page 195, was published a review of the book "Centrifugal Pumps, Their Design and Construction," by L. C. Loewenstein and C. P. Crissey, which I consider so far fails to do the work justice that I request the publication of my views, in the hope that it may do justice to what I feel sure is the best technical book on centrifugal pumps. I believe I can speak with some authority on the subject as I have been identified with centrifugal-pump design for over 20 years and designed for Sulzer Brothers, of Switzerland, some of the very earliest high-pressure centrifugal pumps the success of which really led to the introduction of this type of pump in Europe. Since then I have been identified with several of the most prominent pump builders of England as chief designer and chief engineer.

The reviewer states that "The theory expounded in this book is totally unsuited to the present needs of the pump designer." This is absolutely incorrect; this theory is recognized by the vast majority of pump designers as best suited to their work. It is not suited to any designer who does not believe in the efficiency of diffuser vanes.

The reviewer criticizes the solution of the problem on page 126 but does not state how he would treat the same example. There are, of course, two ways of building pumps; the one he suggests: "A stock of standard (?) pump-casing patterns, each capable of accommodating a limited series of pump diameters, and for each proposition make a special impeller," as a result of which "Eventually the manufacturer accumulates a stock of impeller patterns whose characteristics have been well tested and which are classified." The other way, as outlined in the book, is the more scientific one and also the more economical way; that is, to build the pump impellers with predetermined characteristics and predetermined classification so that the range of possible service of each pattern is well known in advance of actual construction, permitting the designer to cover the possible range of usefulness of each design most economically.

The reviewer states: "The authors believe strongly in the efficiency of guide vanes surrounding the periphery of the impeller"; and intimates that centrifugal pumps without guide vanes are preferable. In fact, he quotes the authors as agreeing to his views when they state on page 83: "By exact and very careful construction, pumps without guide vanes can be made to give fairly satisfactory efficiencies."

The reviewer also tries to prove his point by referring to Fig. 267, which shows a five-stage turbine-driven pump without diffuser vanes which gave at the official tests an efficiency of 58 per cent. The best answer to his views is the fact that of about 30 of the most prominent manufacturers of centrifugal pumps, all described in the book, only one, or possibly two, do not use diffuser vanes, whereas all the others do use them. In fact, the reviewer states that this portion of the book "is exceedingly interesting and gives a very good idea of the present state of the art both in Europe and in this country." If this is true, the present state of the art surely shows most conclusively that diffuser vanes are considered preferable for high-pressure centrifugal pumps and that almost all designers agree with the authors as to their efficiency. Hence, although the book expounds a theory totally unsuited to the needs of those few designers who do not use diffuser vanes, the large majority of designers, engineers and students will find the book admirably adapted to their needs.

One of the best features of the book, and to me the most valuable, was not mentioned. The book is worth its publication if for no other reason than the most thorough and excellent presentation of the theory and calculation of critical speeds and the strengths of impellers. The question of critical speed is of vital importance to designers of high-speed machinery, and until the publication of this book no easy method of computation had ever been published. The theory has been given in an abstract manner by others, but no presentation has ever been so clear and useful as the one given by the authors of this book.

In my opinion, which I am sure is shared by many others, the book is by far the best presentation of the theory and practice of centrifugal-pump design ever published.

HUGO Y. ANGSTROM.

Manchester, England.

Sizes of Turbine Steam and Exhaust Pipes

W. J. A. London, in the July 25 issue, refers to some criticisms of Mr. Neilson, and in return criticizes Mr. Neilson's formula from a practical point of view.

In support of Mr. Neilson I would say that I have had considerable experience in designing exhaust pipes and passages based on his formula, and have found it quite simple to use and perfectly satisfactory in practice. In fact, the curves published in POWER for July 13, 1909, over my name, are based on his formula.

Mr. London refers to sharp bends in pipe lines, but it is obvious that the number of these appearing in a properly designed system would be so small as to have practically no serious defect.

W. VINCENT TREEBY.

Goodmayer, Essex, England.

Central Station Methods

The following, which is the substance of an advertisement which appeared in the daily papers in a certain city, is typical of the distorted and exaggerated statements constantly being made to the public:

"Another Large Private Plant Displaced by Central Station Power.

"After the most searching investigations and thorough practical tests the Hotel —— has abandoned its generating equipment and contracted for our power.

"This is a most notable demonstration of the superiority of central-station service as this isolated plant was one of the largest and most efficient in ——.

"It furnished light and power for the hotel; the —— and Auditorium theaters; Turkish baths and rathskeller. The installation comprises over 100 horsepower in motors; 6753 incandescent lamps, 5 large electric signs and 8 flaming arcs.

"Here are some of the advantages derived from the new power source: A distinct monetary saving, better light, less vibration, less noise, elimination of the excessive heat from the boilers, no dirt, ashes or smoke and the hotel will be from 5 to 15 degrees cooler during the summer than heretofore.

"Our industrial power department will make an investigation of your plant and submit comparative figures for your inspection. It may be the means of saving you a great deal of annoyance and money. Get in touch with us today.

"The Gas and Electric Company."

Here is what the plant actually contained:

Poorly arranged boilers in a room not properly ventilated. A large ventilating fan driven by a piston-valve throttling engine which had no jacketing and which had been allowed to run for probably eight years without any overhauling. An ammonia compressor driven by a common throttling engine without jacketing. A common steam pump with cylinders but poorly jacketed, doing duty as an elevator pump. Four dynamos driven by balanced plug-valve engines which leaked steam at a scandalous rate.

With such an equipment it is not difficult for the central station to displace the plant. But the story would be far different had the plant been rightly laid out in the beginning.

C. R. MCGAHEY.

Baltimore, Md.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Safety Valve Spring Formulas

Will you explain why I cannot make the accompanying statements and formulas for safety-valve springs agree?

The size of steel for safety-valve springs may be found by the following formula:

$$\sqrt[3]{\frac{s \times D}{C}} = d$$

where

s = Load on the spring in pounds;
D = Diameter of the spring in inches from center to center of the wire;
d = Diameter or side of square of the wire in inches;
C = 8000 for round and 11,000 for square steel.

The pressure or load on a safety valve due to the spring may be found by the following formula:

$$\frac{d^3 \times 2}{D} = S$$

where

d = Diameter of the wire in sixteenths of an inch;
D = Diameter of the spring in inches from center to center of the wire;
S = Load on the spring in pounds.

H. D. M.

The formulas are practically identical. The first can be expressed more simply as

$$\frac{S \times D}{8000} = d^3$$

In this formula d is the diameter of the wire in inches. In the second, the same letter represents the diameter of the wire in sixteenths of an inch. The expression for the same quantity will therefore be 16 times as large in the second case as in the first and becomes

$$(16\ d)^3$$

Both sides of the equation are practically multiplied by 16^3 and it becomes

$$\frac{S \times D \times 16^3}{8000} = 16^3\ d^3$$

But 16^3 is 4096, and substituted for the 16^3 in the formula it will be seen that it will go into the 8000 in the denominator very nearly two times, making it read:

$$\frac{S \times D \times 4096}{8000} = 16^3\ d^3$$

$$2$$

Transposing for S it takes the form

$$\frac{16^3\ d^3 \times 2}{D} = S$$

which is the second formula, the 16^3 in the numerator being implied by the fact that d is given in sixteenths of an inch and that these are cubed. If the example is worked out by the two rules, it will come out with only the slight difference due to the failure of 4096 to divide exactly into 8000.

Cutoff Slide Valve Engine

How early in the stroke can a slide valve be made to cut off?

L. L. S.

Five-eighths of the stroke is as early as it is practicable to cut off in a slide-valve engine, though any cutoff desired may be had by adding lap to the valve, but it is impossible to get a full port opening with a cutoff earlier than half stroke.

Horsepower of Vertical Boiler

What is the heating surface in a vertical fire-tube boiler, and what is the horsepower of a vertical boiler with a 40-inch firebox 26 inches high and with forty-two 1½-inch tubes 66 inches long? The water line is 50 inches above the crown sheet.

H. P. B.

The heating surface of a boiler is all of that surface which is exposed to heat upon one side and water upon the other.

In a vertical boiler this surface consists of the inside area of all of the tubes from the crown sheet to the water line; the area of the side sheet of the firebox and the area of the crown sheet minus the cross-sectional area of all the tubes.

It takes 2.87 feet of a 1½-inch tube to make 1 square foot of area, and in 42 tubes submerged 50 inches or 4.166 feet the heating surface is

$$\frac{4.166 \times 42}{2.87} = 60.9 \textit{ square feet}$$

The diameter of the firebox is 40 inches by 26 inches high and has an area of

$$\frac{40 \times 3.1416 \times 26}{144} = 22.68 \textit{ square feet}$$

The inside area of forty-two 1½-inch tubes is

$$42 \times 1.39 = 58.38 \textit{ square inches}$$

The heating surface of the 40-inch crown is

$$\frac{(40 \times 40 \times 0.7854) - 58.38}{144} =$$

$$8.32 \textit{ square feet}$$

Allowing 10 square feet of heating surface for horsepower, the rating of the boiler is

$$\frac{60.9 + 22.68 + 8.32}{10} = 9.19 \textit{ horsepower}$$

Graphite in Boilers

I have been told that graphite will prevent scale from forming in a boiler. Will its use hurt the boiler? Will the steam be so affected that it cannot be used for brewing?

A. J. L.

Graphite will not affect a boiler prejudicially. It is sometimes applied in the form of paint on the inside of boilers to prevent the adhesion of scale to the surface. The steam from such boilers may be used for brewing purposes provided that the boiler does not foam or prime, as in such cases water will be carried over with the steam, and if it contains suspended graphite or other matter it will not be suitable for brewing.

Heating Surface and Boiler Horsepower

How are the heating surface and horsepower of horizontal return-tubular boilers found?

J. F. M.

To find the heating surface of a return-tubular boiler add together the area of one-half of the shell, the area of all the tubes and the area of one head. From this sum subtract twice the area of one end of all the tubes. The remainder will be the heating surface.

For example, the heating surface of a boiler 66 inches by 17 feet with 70 four-inch tubes is

½ shell	146.86 square feet
Seventy 4-in. by 17-ft. tubes	1166.66 square feet
One head	23.76 square feet
Making a total of	1337.28 square feet
From this subtract the area of 140 4-in. tube ends	12.20 square feet
The remainder will be	1325.08 square feet

Ten square feet of heating surface is the standard allowance for a boiler horsepower.

$$\frac{1325.08}{10} = 132.5 \textit{ horsepower}$$

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Contents

PAGE

Developments in Prime Movers..... 392
Steam Engine Lubrication..... 396
Removing Emulsified Oil from Water.... 397
Turbines for Japanese Navy..... 398
Special Crane for Coal..... 398
A High Speed Cast Iron Flywheel..... 399
Why Jimmy was Refused a License..... 401
Heat Transmission in Boilers..... 402
Wooden vs. Steel Boilers..... 403
Wooden Knockoff Plate..... 403
Gas Engine Cycles..... 404
A Bad Wreck from a Small Cause..... 406
Practical Points on Electric Wiring..... 407
Practical Letters:
Cylinder Oil for Hot Bearings.... Turbine Oiling Trouble....Condemns License Law....Straightening a Connecting Rod....Belt Ran to Side of Pulley....Improved Stop Valve.... Daily Log Sheet....No Relief Valve on Heater....Isolated Plant Practice....A Pump Experience....Questions for Discussion.... Prevent Standpipe Freezing.....409-411
Discussion Letters:
Direction of Compressor Rotation ....Filling Oil Storage Tank....Silent Running Engines....Steam in Cold Boiler....Central Station versus Isolated Plant....Lifting Water in Boilers....Trouble with Leaking Tubes....Riverton Turbine Accident .... Locating Keyway in Corliss Valve Stems....Centrifugal Pumps, Their Design and Construction.... Sizes of Turbine Steam and Exhaust Pipes.... Central Station Methods 412-415
Editorials 417-418
Economics of Hot-water Heating..... 419

POWER

Sudden Release of Pressure

A correspondent is doubtful as to what effect a sudden release of steam and water under high pressure will have on a steam boiler. Although this question has been settled by the majority of engineers, there are some who feel convinced that there is but slight danger from such an occurrence.

There is extreme danger from a sudden release of steam, and water either from a return-tubular boiler or a water-tube boiler. When the shell of a return-tubular boiler ruptures it is destroyed by the sudden release of the energy contained in all of the water and steam confined within. The same thing happens when the steam drum of a water-tube boiler ruptures as it contains large quantities of water and steam.

Even the bursting of a steam header may cause an explosion. No better illustration can be cited than the St. Louis explosion, several years ago, when six out of seven water-tube boilers exploded in rapid succession from this cause.

Explosions of this nature are made possible as, although the headers or drum do not contain large volumes of steam, they are connected with tubes which contain enough water and steam under high pressure to blow the boiler to pieces when an initial rupture occurs.

Constant contact with danger makes men careless; they take a chance here and a chance there and are lulled into a sense of security because an accident has never happened to them before. It may be known that a steam header is weak, but these men take a chance that it will hold and nothing is done.

Often the engineer realizes the chances that are taken but he is powerless to have the matter remedied, as is instanced in the case of a certain steam plant installed in an office building about fourteen years ago and in operation today. Water-tube boilers are used and a steam pressure of one hundred and fifty-five pounds per square inch is carried on the boilers and the steam mains. These mains are constructed of standard pipe and fittings, whereas they should be of extra-heavy material.

The chief engineer has always been fearful of these pipes and fittings, especially the latter, but he cannot persuade the manager of the building to install a heavy system.

These pipe lines may continue in service indefinitely or they may be responsible for a boiler explosion at any moment. Thus a chance is being taken, lives are imperiled, and all for the purpose of saving a few dollars.

It is the duty of every engineer to remedy any weakness known to exist in his plant if it is in his power to do so. The bursting of a steam header may result in nothing more than lost steam, and there is grave danger that it may cause a boiler explosion and loss of life.

The Institute of Operating Engineers

The organization of the Institute is an accomplished fact. The first convention has been held and articles of incorporation taken out. For an organization which has, as yet, few local branches and of which few of the members are so situated as to be able to attend a meeting in a distant city at their own expense, the attendance was good, and the interest and enthusiasm inspiring.

A half day sufficed for all the business, including the election of officers, and the remainder of the time was devoted to the presentation and discussion of professional and technical papers. Some of the older societies, which spend the greater part of a week electing a couple of officers, might take a leaf from the new Institute's book in this respect.

Permanent headquarters will be maintained at the Engineering Societies building in New York and engineers, especially those who can qualify as master operating engineers, are invited and urged to connect themselves with it directly.

In the meantime, efforts will be directed at the organization of branches which shall organize local facilities for carrying out courses of instruction enabling the members to meet the requirements of the Institute for the various grades.

The Institute offers the best means yet proposed for a man who must work out his own education and destiny to do so under competent guidance, with the incentive of the fellowship of co-workers in the same lines and with the assurance of recognition of such work as he shall accomplish and such merit as he shall develop. An inquiry addressed to the secretary of the Institute of Operating Engineers, Engineering Societies building, 29 West Thirty-ninth street, New York, will bring full particulars.

Pulleys and Belting

It is often loosely said, when speaking of the behavior of belting, that "a belt will always run to the high side of the pulley." Without concise definition the term "high side" is a meaningless expression. A belt running over a pulley will always tend to move laterally toward that portion of the pulley face with which it first comes in contact, regardless of the contour of the face of the pulley, its alinement or that of the shaft. It is the direction of the belt going onto the pulley that decides how it will run and not the direction in which it leaves. Shafts are parallel when their center lines are the same distance apart throughout their length. Pulleys on parallel shafts are in line when a line joining the middle of their faces is square with the shafts, and when on shafts not in the same plane they are in line if a line from the middle of the face of the pulley where the belt leaves falls on the middle of the face of the receiving pulley at right angles with its axis. Both pulleys and shafts may be so far out of line as to wholly overcome the tendency of the belt to move toward the portion of the pulley first touched, but this fact does not destroy the tendency. It overcomes it.

If a pulley has a crowned face, the middle of the crown is the portion first to come in contact with the belt and it will tend to keep it running on the middle of the pulley face, which is in this case the high side. Where it has a straight face and is much wider than the belt, the belt will run equally well on any part of the pulley face if the shaft is in line. If the shaft is out of line, that portion of the pulley rim which is furthest from the other shaft is the high side, but the belt will run away from this side toward the other or low side because the first contact is on this portion of the pulley face and the belt must move toward it.

It will not require a great deal of study on the part of anyone to get a thorough understanding of the governing principles in this matter as the behavior of a belt running onto a pulley is identical with that of a beam moving on rolls.

If the beam is square with the rolls it will move only in the direction of its length. If, however, one of the rolls is at an angle with the beam it will in turning move sidewise as well as forward and the lateral movement will be toward the end of the roll where the first contact comes and which, if the roll were a pulley and the beam a belt, would be the low side.

It is the direction in which the belt goes onto the pulley that determines its behavior. The direction of leaving is a matter of no consequence whatever. It is this that makes the running of crossed, quarter-turn and any angle belts possible. So long as a belt goes onto a pulley at right angles to its axis it will run satisfactorily, regardless of high or low sides.

Oil in Boilers

When the boiler-feed water is taken from the hotwell of a jet condenser, as is commonly the practice where fresh water is available for condensing purposes, there is little danger that excessive quantities of oil will be carried to the boiler. There is believed to be little danger to any type of boiler from the oil itself because, being lighter than the water, it will not sink to the bottom and become attached to the plates. The error in this belief lies in the assumption that what is oil when it enters the boiler remains oil. It does not. Subjected to the long continued high temperature of the water, it is slowly distilled, that portion which is volatilized going off with the steam, leaving behind a residue of a higher specific gravity than water, which sinks to the bottom as soon as the circulation stops. When cold it is firm and nonadhesive, but when heated is not unlike asphaltum.

So long as the circulation continues, this matter will travel along with the current and do no harm. When it settles on the comparatively warm sheet it adheres to it so strongly that it is not dislodged when the circulation starts again. Water is kept from contact with the sheet and a "bag" is the result.

When a bag is caused by oil it is liable to be anywhere on the shell below the fire line. It is usually near the bottom, but may be well up on the side and anywhere between the front and rear heads.

The mass that clings to the sheet is not always entirely oil residual. It is often found agglomerated with the scale-making solids of the water in irregular masses in all parts of the boiler.

What is really more dangerous, because less liable to be suspected, is the milky-colored emulsion which goes to the boiler after the heavier-bodied oil has been removed by filtration. This does not behave like oil proper in associating with foreign matter nor by gathering in large or small masses, but goes atom by atom to the shell of the boiler, which it covers with a tissue of hard, bright, black varnish, which as effectually prevents the contact of water as the thickest coat of scale. It covers the entire sheet below the fire line before it is noticed and the "bag" is often as wide as the diameter of the boiler and as long as the sheet. This bagging is gradual, as at the first slight stretch the varnish cracks and the water cools the softened sheet and restores its strength until the coat of varnish again covers it; the operation is repeated until the sheets are bulged like a barrel. Neither the bulging nor the bagging can be avoided without completely removing the oil and its emulsion from the water.

Clean, or Pretty Clean?

Are your boilers really clean, or do you only think that they are? This thought is suggested by the inside history which we have just learned of the placing of a tube cleaner.

The cleaner was one of the "knocker" type and the boilers were horizontal tubulars. The manager of the plant was not at all convinced of his need for such an apparatus. The water was first-class, they cleaned out frequently, the boiler inspector complimented them on the excellent condition of their boilers and there seemed to be but little chance that they would buy a cleaner, but the manager consented to let the salesman send one on trial, to be kept and paid for only if it proved so satisfactory that he really thought they wanted it.

The cleaner was sent and tried, but it removed no scale; instead, it disclosed considerable "crust" on the inside or fire side of the tubes. The manager and his engineer then concluded that what they wanted was a soot and crust remover. There was, he said, practically no scale in the boiler except at the back ends of the tubes, and this the engineer preferred to remove with a hammer or chain.

The manufacturer of the cleaner pleaded for a fair trial as the cleaner depended upon the vibration set up in the tube to free the scale and it could not get up this vibration when pounding upon a cushion of soot. He pointed out that by the use of the head provided for that purpose the apparatus itself would clean the inside of the tube, and then by the substitution of the hammer head it would knock the scale off from the outside. The manager consented to allow the cleaner to remain until another opportunity offered to try it. When it came he wrote that after thoroughly cleaning the tubes of soot the cleaner had been put through two boilers, and over four hundred pounds of scale were taken out, some of which was more than a quarter of an inch in thickness. The boilers had been washed out thoroughly twice a month, the visible surfaces appeared to be clean, and they were amazed to find that there was any scale to speak of.

It appears that this is not an uncommon experience. A boiler is "pretty clean"; it keeps on steaming all right, the inspector is satisfied and nothing more is done about it, whereas a good, thorough cleaning will reveal unsuspected accumulations, improve the efficiency and prolong the life of the boiler.

On August 24, a cylinder head blew out at the Greylock Woolen Mills, North Adams, Mass. It is reported that the engineer was badly injured and a number of the workers hurt. It is also stated that the building was badly damaged.

Heating and Ventilation

Considered as power plant problems. Layout and operation of systems and apparatus

Economics of Hot Water Heating*

By Ira N. Evans†

The following article is intended to give some actual results in the form of curves and tables from a plant installed as described in the July 18 number, and also a method of getting at the coal consumption and the cost of exhaust heating applicable to any plant.

Most owners and manufacturers can tell to a small fraction the amount of fuel required for power and will spend large sums to compound or condense an engine for a comparatively small saving. But when it comes to the heating system no records are kept, and they know little or nothing as to the actual fuel cost of operation. They generally assume that if they are utilizing the exhaust steam the heating costs nothing. When it is decided to extend a plant or build a new one the engines and boilers are purchased first without reference to the method of heating or probable heating load beyond the maximum requirements in zero weather. Later, the heating plant is modified to meet cuts in the original appropriation.

The type of heating system, especially if hot water is under consideration, should be just as much a factor in determining the size and type of engine and the arrangement of the plant as are the power load and physical location.

In many cases where it is proposed to use exhaust steam for heating, the plant is actually so arranged as to furnish an excessive amount and the heating apparatus is then made uneconomical in utilizing this steam because there is a surplus, notwithstanding the fact that the heating system is only operated seven months in the year. This, of course, would not be acknowledged in any particular case but instances have occurred where this is the final result.

The diagram, Fig. 1, from the previous article and reproduced herewith, shows a plant arrangement for a hot-water heating system using forced circulation, which does not interfere with the arrangement or the operation of the condenser and engine at any time and yet the heating system utilizes the exhaust steam from the turbine under partial vacuum.

By this arrangement the heating system reduces the work on the regular condenser by performing its function to the extent only of the actual steam requirements for heating at any given outside temperature condition.

The system will condense all the steam at any vacuum regardless of the variation in the engine load, and the temperature of the water in the heating system can be varied at will to meet outside weather conditions by changing the vacuum. With a constant temperature in the space heated, as the weather grows colder the condensing capacity of the heating system increases in proportion to the decreased vacuum and increased steam consumption on the turbine.

Fig. 2 shows the curves of operation of the plant of the Lackawanna railroad at its Hoboken terminal, arranged substantially as shown in Fig. 1. These curves were obtained by keeping a log to determine the requirements of water temperatures for satisfactory heating, and the plant is at present operated according to this schedule. In this plant there are about 70,000 square feet of radiation and the water circuit is about a mile in length.

There are several buildings. The power plant, centrally located at the end of the train shed, consists of about 3000 horsepower in boilers, two large compound Ingersoll-Sergeant air compressors for furnishing air for the signals and switching system in the yard as well as the air pressure needed to adjust the water level in the expansion tank *T* of the heating system. The amount of water and air in this tank are regulated in conjunction with the differential gages which show the static head on the system. The tank is kept about 25 per cent. full of water, the air pressure being used to lower the level of water in the tank by forcing it into the system. An automatic air trap releases the displaced air at the top of the system by opening when no water is present and closing when the water raises the float.

There are two Westinghouse-Parsons turbines of 500 kilowatts capacity each for furnishing light and power in the buildings. The power for operating the motor pump on the heating system is taken from either of these turbines. The

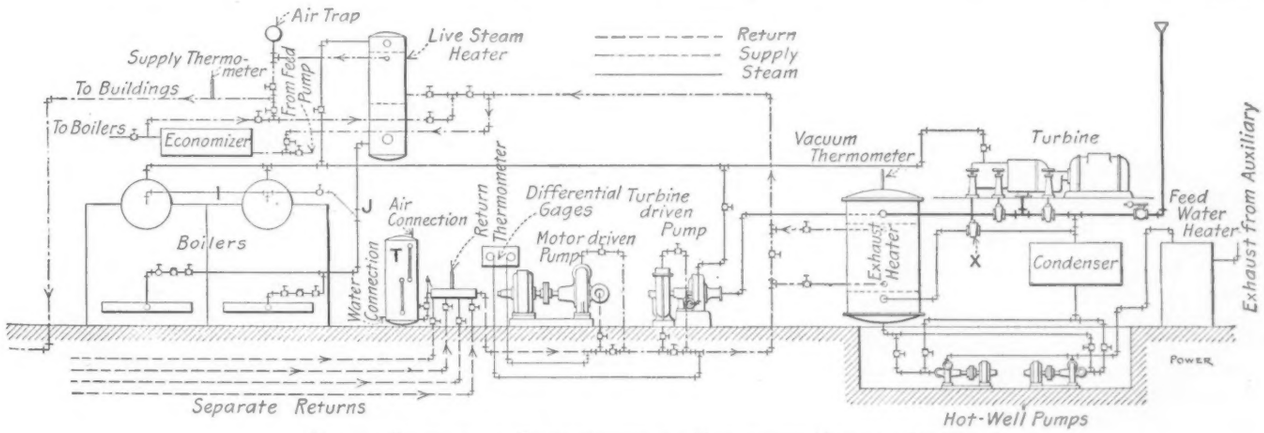

Fig. 1. Diagram of Connections of Hot-water Heating System

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economizer is not connected to the heating system as the plant operates 24 hours a day and the waste heat from the gases is always economically utilized for feed purposes.

The plant is operated condensing, taking water from the North river, and the heaters and pumps for the hot-water heating system are so arranged as to utilize the exhaust steam under partial vacuum from either machine, as either

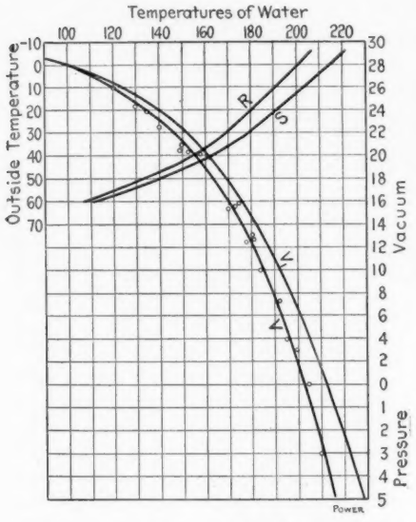

FIG. 2. TEMPERATURE-VACUUM CURVES

turbine may be operated on either condenser with full vacuum by manipulating the floor-stand valves without stopping the units.

As may be seen in Fig. 1, pressure gages on either side of the pumps show by their difference in reading the pounds pressure of friction head against which the pump is operating, and thus indicate the rapidity of circulation. The thermometer on the steam chamber of the exhaust heater shows the temperature of the reduced vacuum required for any given water temperatures indicated by the thermometers on the return header and supply main. The difference in readings of these two thermometers is the drop in temperature or the number of degrees absorbed in the heaters and extracted by the radiation. On this particular job, recording instruments are used throughout and charts are inspected daily by the chief for any unusual developments.

Curves *R* and *S*, Fig. 2, give the readings of the supply and return thermometers plotted as abscissas with the corresponding outside temperature as ordinates. Curve V_1 shows the theoretical vacuum in inches and pounds as ordinates corresponding to the supply temperature as abscissa. Curve *V* is the actual vacuum indicated by the thermometer on the exhaust heater with the readings plotted as ordinates and the corresponding supply-water temperatures as abscissas.

By following the outside temperature lines to their intersection with the curves *R* and *S*, the corresponding return and supply-water temperatures are found, and by following the water-temperature lines to their intersection with V_1 and *V*, the theoretical and actual vacuum can be read from the corresponding figures on the right side of the chart. When the weather becomes cold the vacuum curve is steep and a slight increase in water temperature requires a correspondingly greater decrease in vacuum or increase in back pressure.

The shape of the curves *R* and *S* depend entirely on the amount of direct radiation and air supply; indirect heating may require a higher temperature of water in extreme weather and a correspondingly lower vacuum.

The amount of water circulated is found to be nearly constant for a given speed of the pump and was obtained by a recording instrument on a venturi meter placed in the water circuit. The amount at present is 3,350,000 gallons in 24 hours or 1,158,700 pounds per hour. The amount of water and the speed of the pump are not varied, the reduction in heat transmission being accomplished entirely by changing the temperature of the circulating water and varying the vacuum. This mistake of varying the water circulation and causing a higher average water temperature has been made in a number of cases.

In extreme weather, due to the rapid decrease in vacuum, the advantages of connecting the circulating pumps in series and operating both become apparent as the increased circulation reduces the drop and the average water temperature, causing greater efficiency of transmission in the heaters and radiation and reducing steam consumption.

Table 1 gives the data from which Fig. 3 was plotted. The supply and return temperatures with corresponding vacuums were taken from Fig. 2 for each 5-degree interval of outside temperature. The vacuum in each case is the maximum that can be carried to give the proper water temperature to heat the spaces properly for any outside corresponding temperature.

Curve *H* is obtained by plotting the pounds of steam per hour given under *H* in the table with the corresponding outside temperature.

Curve *T* shows the amount of exhaust steam available under different vacuums with a constant load of 400 kilowatts at the switchboard. The different rates are given in Table 1 under the proper heading and total number of pounds of steam is found by multiplying by 400. These values are given under the heading "Curve T," and are plotted in conjunction with the corresponding vacuums and outside temperatures.

Curve *R* is the steam consumption of a 650-horsepower compound-condensing

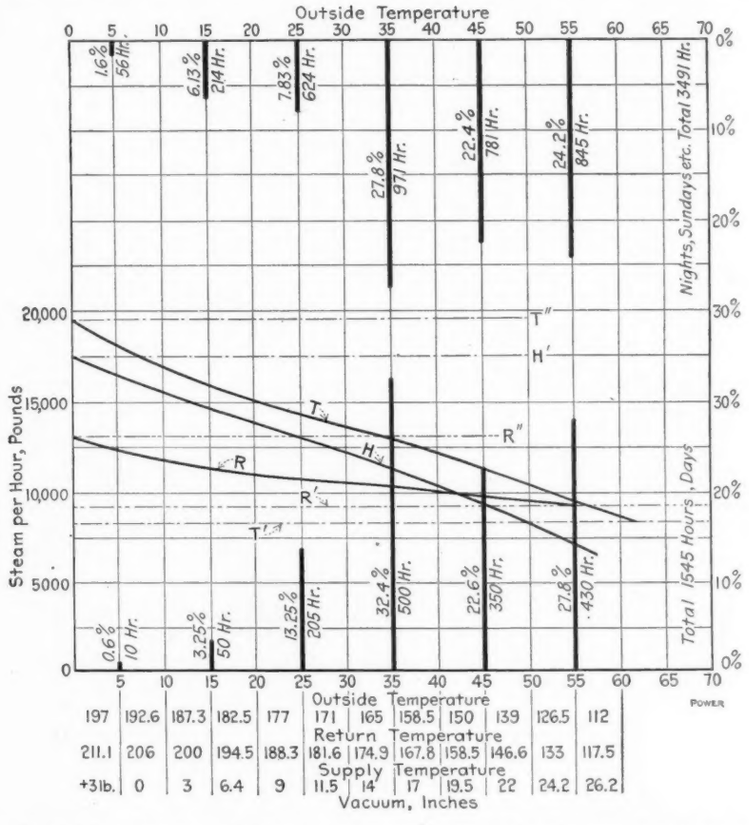

FIG. 3. CURVES SHOWING RELATION BETWEEN STEAM CONSUMPTION OF TURBINE AND ENGINE AND THE REQUIREMENTS OF HOT-WATER HEATING SYSTEM

engine with 100 per cent. load and is assumed as the nearest reciprocating unit in order that its results may be compared with the turbine. The rate and total steam consumption under the different vacuums are given in Table 1. Line H', Fig. 3, indicates the steam consumption of a heating system with the medium at 210 degrees and with all heating surface turned on. Line T'' is the steam consumption per hour of the turbine with 400 kilowatts load and slight back pressure. Line T' represents the steam consumption with the turbine under 28 inches of vacuum. Line R' is the steam consumption of the reciprocating engine with 400 kilowatts load at the switchboard and a vacuum of 26 inches. R'' indicates the steam consumption of the reciprocating engine with a load of 400 kilowatts at the switchboard, or 650 indicated horsepower, under a slight back pressure. By subtracting the pounds of steam available at a vacuum of 28 inches from the quantities corresponding to the various vacuums given in Table 1, the column under $T - T'$ is obtained. These figures indicate the cost in pounds of exhaust steam of utilizing the exhaust steam for the heating system and running the turbine under partial vacuum. $T - H$ is the excess steam in any case available for further additions to the heating system.

Particular attention is called to the fact that the curves T and H are similar for nearly their entire length and that with a constant load by changing the vacuum the proper steam supply at the proper temperature is available to heat the circulating water of the heating system.

The turbine load can be decreased slightly or more surface added to the heating system and the curves T and H will nearly coincide. The exact figures are given in column $T - H$, Table 1. The minimum difference is 1127 pounds and the maximum 2995 pounds, but in the latter case, by observing curve H, Fig. 3, it may be noticed that the line goes below the requirements of the turbine on full vacuum, or curve T', and the actual cost $T - T'$ is only 960 pounds.

The cost of operating the reciprocating engine is not worked out in detail, but it can readily be judged from the curves. The curve of steam consumption under different vacuums does not coincide at all with curve H. The engine would operate at no vacuum until R'' crosses H at 25 degrees outside temperature and there would be only about two changes in vacuum available between 25 and 60 degrees outside temperature, or full vacuum. All periods colder than 25 degrees would require the addition of live steam or another unit operated noncondensing. In the latter case the friction load and losses of the additional machine would nearly cancel any saving over using live steam.

The large reciprocating engine does not lend itself readily either to wide variation in load or wide range of vacuum as the valve setting and receiver pressure have to be readjusted.

Considering the flexibility of the turbine for change in load and vacuum, ease of operation, floor space, and slightly better steam consumption for overloads and full vacuum, the economy of the turbine is fully as good, if not better, than the reciprocating engine for exhaust hot-water heating with partial vacuum on units of over 250 kilowatts. Engineers in general have been reluctant to reduce the vacuum on large turbine units on account of the rapid increase in steam consumption, but a study of the arrangement in Fig. 1 and the charts will show the advantages to be derived from this practice.

In making the comparison of the turbine and reciprocating engine the number of hours operation (about 1500) in summer when no heating is required should be taken into account when the steam consumption of the reciprocating engine under the same load and full vacuum will be greater than that of the turbine. In this case it would be

9100 — 8240 = 860 *pounds per hour*

The steam consumption on the reciprocating engine increases with both reduction and increase of load on the same cut-off, while the turbine rate decreases with increased load.

Another correction which in general should be applied to the steam consumption of the engine, when the exhaust is used as described, is the necessary steam to heat the engine condensation from the temperature of the vacuum in each case to 200 degrees. At full vacuum the raise in temperature would be about 100 degrees. At atmosphere it would be zero. This reduces the actual cost of operation $T - T'$ by the amount of auxiliary steam necessary to be used in the feed-water heater. This steam should be added to the total steam consumption for each change of vacuum to get the true cost. This has not been considered in this discussion on account of complicating the subject.

Up to the present, the only method of using the exhaust from turbines for heating has been under atmospheric or slight back pressure, or at a rate of about 50 pounds per kilowatt-hour. With the arrangement shown in Fig. 1 the steam consumption is reduced to about 30 pounds average, or less, for all of the steam used in the heating system.

The steam consumption of any heating system when all of its surface is turned on is absolutely fixed by the temperature of the pipes and the temperature of the room. The steam consumption of the turbine or engine is absolutely fixed by the load and degree of vacuum or back pressure whether it is required in the heating system or not. The only alternative in a steam-heating system or vacuum system where the steam cannot get below 212 degrees is to open the windows or shut off the radiators. The former course is the one generally pursued, and with the rise in tempera-

TABLE 1. DATA FOR CURVES H-T-R OF FIG. 3

		S	R	D	V	L	Curve H		Curve T		Curve R	$T - T'$	$T - H$
Outside Temp. Deg. F.		Supply Therm. Deg. F.	Return Therm. Deg. F.	Dif. Temp. Deg. F.	Max. Vac.	Latent Heat.* B.t.u.	Lb. Steam per Hour. $\frac{1158700}{L} \times D$	400 Kw. T Rate	Lb. Steam T 400 Kw. Load	650 H.P. R Rate	Lb. Steam R 400 Kw. 650 H.P.	Cost T Operated Red. Vac.	Surplus Steam T
1	0– 5	211.1	197	14.1	3 lb.	960	17,020	47	18,800	19.5	12,675	10,560	1,780
2	5–10	206	192.6	13.4	0	966	16,073	43	17,200	18.5	12,025	8,960	1,127
3	10–15	200	187.3	12.7	3″	969	15,187	41	16,400	18–	11,700	8,160	1,213
4	15–20	194.5	182.5	12	6.4″	973	14,292	38.8	15,520	17.4	11,310	7,280	1,228
5	20–25	188.3	177–	11.3	9″	977	13,402	37	14,800	17–	11,050	6,560	1,398
6	25–30	181.6	171–	10.6	11.5″	981	12,520	35.2	14,080	16.6	10,790	5,840	1,560
7	30–35	174.9	165	9.9	14″	986	11,654	33.5	13,400	16.2	10,530	5,160	1,746
8	35–40	167.8	158.5	9.3	17″	993	10,853	31.4	12,560	15.6	10,070	4,320	1,707
9	40–45	158.5	150	8.5	19.5″	1,000	9,851	29.5	11,800	15.1	9,815	3,560	1,949
10	45–50	146.6	139	7.6	22″	1,007	8,745	27.3	10,920	14.6	9,490	2,680	2,175
11	50–55	133	126.5	6.5	24.2″	1,016	7,412	25.2	10,080	14.3	9,295	1,840	2,668
12	55–60	117.5	112	5.5	26.2″	1,027	6,205	23	9,200	14	9,100	960	2,995
13					28″			20.6	8,240				

1,158,700 lb. water circulated per hour.

*Old values used before Marks and Davis tables were adopted.

Fig. 4
- T' full vacuum 1,545 × ordinate T_{13} = 1,545 × 8,240 lb. = 12,730,800.
- R' full vacuum 1,545 × ordinate R_{13} = 1,545 × 9,100 lb. = 14,059,500.
- T'' no vacuum 1,545 × ordinate T_1 = 1,545 × 18,000 lb. = 27,810,000.
- H' steam 210° 1,545 × ordinate H_1 = 1,545 × 17,000 lb. = 26,265,000.

Fig. 5
H' steam 212° 3491 × 17,000 lb. = 59,347,000.

ture of the room the exhaust is taken care of up to the limit set by the relief valve. Some people have claimed that a vacuum could be carried on an ordinary steam job, but this is impossible mechanically, beyond a few inches on account of air leaks in the stuffing boxes, etc.

These leaks increase as the packing gets dry and the vacuum increases. In Fig. 3, *T″* would be the steam consumption of a turbine operated on a heating system at about atmospheric pressure whether the steam was required for heating or not. The surplus would go out through the relief valve. Line *H′* would be the steam consumption of a low-pressure steam system less the amount due to rise in temperature in overheated rooms with all surface turned on. *R″* would be the amount furnished by a reciprocating engine under the same conditions, but if the heating surface was arranged to use the steam at 212 degrees economically in weather 25 to 30 degrees outside temperature, a prohibitive back pressure would be required in zero weather to increase the temperature of the radiation, as well as additional live steam because the radiation is fixed and the only way of varying the supply of heat is by varying the temperature of the steam. This is the reason for poorly heated buildings during sudden cold snaps. It is also the reason why many owners think they are getting the heating for nothing when using the exhaust. The steam from the engine may balance the heating and both be excessive.

The data in regard to the use of the steam for heating in this plant are the result of many months' operation and are absolutely reliable. They show that the heating steam with all surface turned on varies from 6205 pounds at 60 degrees outside temperature to 17,000 pounds in from 0 to 5 degrees weather outside. This shows the main saving in a hot-water plant whether exhaust or live steam is used to heat the water. It also explains why many mills use live steam for heating at high pressure. The cost of the plant and comparatively few hours of exhaust heating (1500 as against 3500 hours of night and holiday heating) makes the proposition one to be considered.

It would be possible to obtain a lower water temperature and greater vacuum by increasing the proportion of heating surface in the building, but the temperatures given have been found about right for the climate in the vicinity of New York City and do not make the cost of the installation excessive.

Supposing the room to have a temperature of 70 degrees and the water 200 degrees average for zero weather, the difference would be 130 degrees. If the temperature of the water were reduced to 185 degrees it would increase the first cost in amount of heating surface

$$\frac{(200-70)-(185-70)}{130}$$

$$= 0.115 = 11.5 \textit{ per cent.}$$

In a cold climate it would be economical to make this increase, but near New York the coldest winter temperature average is about 5 degrees above zero and it is the better practice to maintain the temperatures as given. In general it is best to proportion the heating surface so that no vacuum will be carried in the coldest weather.

In designing a plant of this character experience is necessary in proportioning the heating surfaces of heaters and radiators so as to keep the initial cost down and still operate the engines economically. Excessive back pressure and high temperatures of water will cause the plant to be operated entirely on live steam in colder weather, as the engines will not be capable of carrying the load with back pressure. High velocities in the mains, minimum drop in temperature and ample pumping capacity are absolutely essential for proper operation.

The decrease in steam consumption on the turbine with a 400-kilowatt load between no vacuum and full vacuum is, according to Table 1, 10,560 pounds, and the change in steam requirements on the heating system between 0 and 60 degrees is 10,815 pounds per hour. This shows nearly the same range. It should not be lost sight of that the heating system acts as a condenser, taking a portion of the regular condensing load and reducing the injection water required in the winter time. The curves also make it apparent why the condenser cannot be used as an exhaust heater where a turbine is operated under partial vacuum. This opens up a field for cooling towers and air condensers where heretofore the exhaust-heating problem seven months in the year has eliminated them from consideration.

Although this plant is operated 24 hours a day the writer has used the data in Table 2 to show how the steam consumption could be determined in a plant when the heating nights, Sundays and holidays is operated on live steam due to the main engines being inoperative. This table is very useful in determining the total steam on any heating plant in the New York district. A like chart can be made for any district by obtaining the record of temperature, maximum, minimum and average from the United States Weather Bureau office in that district for each day and month of the heating season. These are plotted on regular thermometer charts, making a day operation of 10 hours and night operation of 14 hours and indicating holidays and Sundays by washed spaces of heavier or lighter color. The weather clerk can usually give a typical daily curve for that district and the minimum and maximum temperatures occur generally at the same hour each day. These temperature charts when pasted together give a continuous curve of temperature for the month with the night and day periods for week days, Sundays and holidays clearly indicated. A heavy line may then be drawn through the average for each 5- or 7-hour period. The total number of hours for each 10-degree period of outside temperature can be obtained and tabulated for use as in Table 2.

In Table 2, *A* shows the engines would be operated in the ordinary industrial plant and exhaust steam available 1545

TABLE 2. NUMBER OF HOURS FOR EACH 10 DEGREE PERIOD OF TEMPERATURE FROM 0 TO 60 DEGREES, NEW YORK DISTRICT

Month	Week Days												Holidays, Sundays											
	"A" Days. 7 a.m. to 5 p.m.						"B" Nights. 5 p.m. to 7 a.m.						"C" Days. 7 a.m. to 5 p.m.						"D" Nights. 5 p.m. to 7 a.m.					
	0°-10°	10°-20°	20°-30°	30°-40°	40°-50°	50°-60°	0°-10°	10°-20°	20°-30°	30°-40°	40°-50°	50°-60°	0°-10°	10°-20°	20°-30°	30°-40°	40°-50°	50°-60°	0°-10°	10°-20°	20°-30°	30°-40°	40°-50°	50°-60°
Oct.				5	35	85					56	140					10	30				28	21	63
Nov.				90	105	20				168	105	14				35	45					119	14	
Dec.		25	100	80	20			70	175	28	14			15	45	10	15			56	70	14		
Jan.		10	60	135	25		28	14	84	161	14			5	25	35	15			7	10	56		
Feb.	10	15	40	95	35	15	14	28	70	105	28	21		5	40	25			14	14	84		14	
Mar.			5	90	90	55			21	168	84	49				5	35	15					42	70
April				5	35	155				14	133	154					10	40					70	42
May.					5	100					28	154						25					28	28
Total	10	50	205	500	350	430	42	112	350	644	462	532		25	110	110	130	110	14	77	164	217	189	203
PerCent.	0.19	0.99	4.07	9.94	6.95	8.54	0.83	2.23	6.95	12.8	9.2	10.6		0.5	2.19	2.19	2.58	2.19	0.27	1.53	3.26	4.3	3.76	4.04
Total	1545 hours 30.68 per cent.						2142 hours 42.6 per cent.						485 hours 9.65 per cent.						864 hours 17.16 per cent.					
	3491 hours 69.32 per cent.																							

hours or 30.68 per cent. of the time and shut down 3491 hours or 69.32 per cent., when live steam would have to be used for heating. In case night operation or two shifts a day were under discussion the night periods would amount to B, or 2142 hours additional.

Many engineers have tried to use monthly averages in figuring heating, but an inspection of these tables shows February with 15 hours at 50 to 60 degrees, and all months have periods at 40 to 50 degrees. With the wide range in heating requirements shown in Fig. 3, a monthly average would not give true results.

In Fig. 3 the heavy lines are drawn across the chart to show graphically the relative time taken up in one year by each temperature period. The heavy lines at the top indicate night and holiday operation ($B + C + D$, Table 2) with live steam; and the heavy lines at the base show the time the engines were operative under different vacuum requirements.

Fig. 4 shows graphically the pounds of steam and the number of hours for each range of temperature. The total length of the chart is 1545 hours, or the total period of engine operation.

Fig. 4 shows graphically by the areas the total pounds of steam required per heating season for each 10-degree period of temperature. The length of the chart is 1545 hours, or the total period of engine operation. The ordinates are the average period taken from curve H, Fig. 3, and from Table 1, multiplied in each case by the number of hours A, Table 2. The area below line H is the sum of these products or the steam required for heating during the day schedule, or 1545 hours per year. The requirements in this particular case were 15,364,260 pounds. The total cost of using the exhaust of the turbine under partial vacuum, $T - T'$, is found in like manner to be 5,818,000 pounds. The lines T'', T', R', R'' H', Fig. 4, were plotted from Table 1 for 1545 hours in each case. R'' is the requirement of a reciprocating engine running noncondensing.

Fig. 5 shows in like manner the pounds of steam required for the heating season, nights, Sundays and holidays, when the engines are inoperative. The total length of the chart is $B + C + D$, Table 2, for each 10-degree period, or 3491 hours. The ordinates for H were taken from Table 2 and Fig. 3, and the total area H, Fig. 5, is 36,102,377 pounds, or the steam required for night and holiday heating by the hot-water system. In Fig. 5, H' is the amount of steam required for 3491 hours with all surface in operation and the medium at 210 degrees. This amounts to 59,347,000 pounds.

Areas H and H', Fig. 5, show the relative economy of hot water for heating versus low-pressure steam with a vacuum system if there were no exhaust steam available, and indicates an increase of about 65 per cent. in steam with all surface in operation for the latter.

If the outside weather were constant in temperature there would be slight difference in economy between the steam and water, but with the daily and hourly variation an economical heating system requires a medium of high specific heat as well as one possible of as wide variation in temperature.

By dividing the pounds of steam by the water evaporated per pound of coal, the number of pounds may be determined and knowing the cost per ton these figures may be readily transformed into dollars.

consumption under full vacuum, or area T', less the excess steam space $T' - H$. It also shows that the greater the economy of the engine under full vacuum the less the apparent saving by the use of exhaust steam for heating. The steam spaces $T - H$ represent the steam going to the condensers. The net saving by operating partially condensing in the case under consideration over using live steam for heating would be

$$\frac{T' - (T - H)}{H}$$

or

$$\frac{12,730,800 - 3,185,140}{15,364,260} = 62 \text{ per cent.}$$

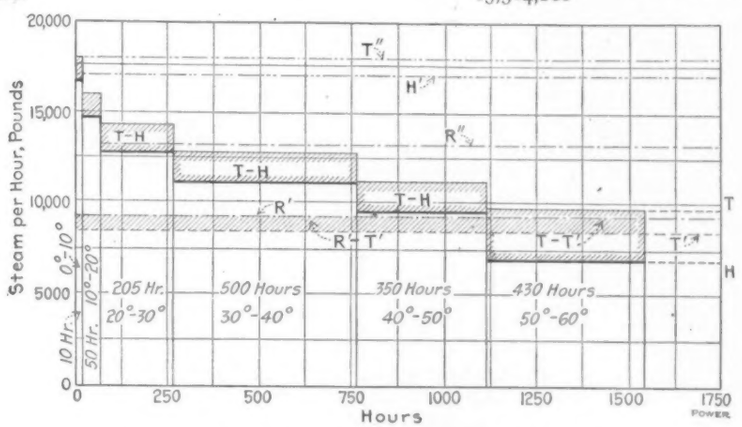

FIG. 4. AREAS GIVING TOTAL STEAM PER SEASON

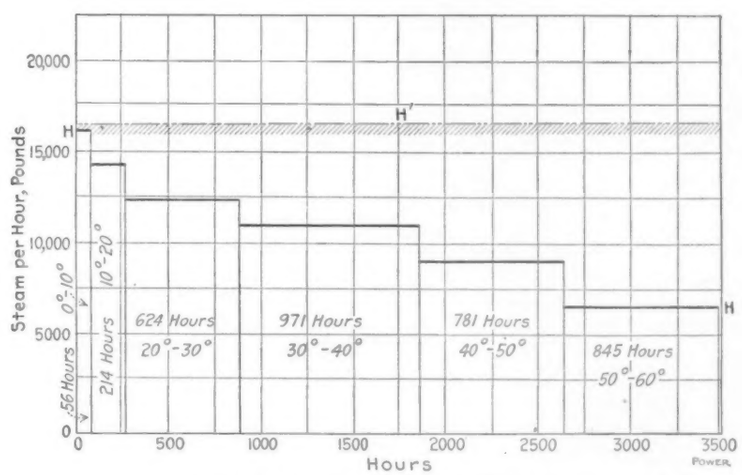

FIG. 5. PERIODS OF LIVE-STEAM OPERATION

The cost per 1000 pounds of steam, allowing $3 per short ton of 2000 pounds for coal, handling ashes, etc., at 9 pounds of evaporation will be

$$\frac{1000 \times 300}{2000 \times 9} = 17 \text{ cents nearly}$$

In showing the amounts represented by the relative areas in Fig. 4, the least cost of heating and power combined, T, was taken as 100 per cent. and tabulated as shown in Table 3.

Fig. 4 illustrates the economy effected by using the exhaust steam under partial vacuum and that it is equal to the steam

The actual amount will be 9,546,600 pounds at 17 cents per 1000 pounds of steam, or $1622 per season of 1545 hours.

The least cost of heating and power, night and day, on a plant operated as described would be, from Table 3, Figs. 4 and 5,

$$\$3153 + \$6140 = \$9293$$

If the turbine were operated at atmosphere and the system on a medium at 210 degrees, or steam at atmosphere, the cost would be T'', Fig. 4, and H', Fig. 5, added, or

$$\$10,090 + \$4588 = \$14,678$$

This is

$$\$14,678 - \$9293 = \$5385$$

an increase of 58 per cent. If the turbine were operated at 400 kilowatts for 1545 hours under partial vacuum, the rate per kilowatt-hour obtained from the exhaust steam used on the heating system would be, from T'', of Table 3,

$$\frac{18,549,000}{400 \times 1545} = 30 \textit{ pounds per kilowatt-hour}$$

The rate per kilowatt-hour, if the exhaust steam were used at atmosphere, would be, from T'', Fig. 4 and Table 3, for the same load and period

$$\frac{27,180,000}{400 \times 1545} = 45 \textit{ pounds per kilowatt-hour}$$

This shows a gain of 50 per cent. in power by using the exhaust steam under partial vacuum, over using it at atmosphere.

The saving by the use of the exhaust steam on a vacuum system shows greater apparently than a hot-water system such as described, but the heating and power combined would be 50 per cent. greater. If the steam for heating balances the power steam the amount can be indefinitely increased and a greater saving apparently shown by the use of exhaust steam for heating.

Many claim that it is unnecessary to operate the heating system continuously when using steam as a medium, especially nights. There is of necessity a surplus of heat on a steam system in average weather (which, it is often claimed, costs nothing, because it is a byproduct), but it is considerable trouble to start circulation and relieve the system of air, and from actual tests it takes more steam to operate a large plant intermittently, allowing the building to cool, than it does to maintain continuous operation.

Automatic heat control is used successfully to reduce the steam consumption and overheating by shutting off the surface, but the first cost and maintenance give the hot-water system the advantage.

In making these comparisons the loss in mains and pumps for circulation has been disregarded, but experience proves that with a properly arranged plant the expense is in favor of the water system.

The reciprocating engine would be less economical in summer by the area $R' - T'$. If the heating surface was constant on a vacuum-steam system and the engine operated at atmosphere, or 210 degrees at the period 30 to 40 degrees, Figs. 3 and 4 show that live steam would be required for the colder periods, and the period 0 to 10 degrees would also require a greater temperature of the medium by 30 degrees, or 10 or 15 pounds back pressure. This might require the engine to be cut out of the heating system and live steam to be used in extreme weather, causing enormous increase in boiler power at the peak of the heating load.

The problem in this article contemplated the constant load of 400 kilowatts, but on the plant in question the load is variable, ranging from 200 kilowatts at noon to 600 kilowatts at night for 24 hours. When a variable load is under consideration, a chart should be worked out for the heating in pounds of steam, as in Fig. 3, and one for the daily load, showing the average hourly variations for 24 hours. With pounds of steam as ordinates and hours as abscissas, the curves of steam consumption for each 5 inches of vacuum between full vacuum and atmosphere should be plotted for the engine or turbine for the average daily load, showing the hourly variations. From these charts one should be constructed for each 10-degree period from 0 to 60 degrees with pounds of steam as ordinates and hours as abscissas. Each chart will have four curves: the typical curve of temperature outside, showing the hourly variation; the turbine steam consumption at full vacuum; the heating curve in pounds of steam, and the curve of actual steam consumption of the turbine at the hourly loads.

By dividing the hours, Table 2, by 24, the number of days can be obtained for each 10-degree period of temperature. When the areas between the curves on each of the six charts are determined and multiplied by the number of days in each case, the heating, excess steam and cost of reducing vacuum can be determined for the season.

When the load is at the peak, one machine may not carry it at reduced vacuum. It is economical then to turn the turbine into the condenser with full vacuum, especially when the peak is of only two or three hours' duration. The circulating pump will continue to operate but no heat will enter the system. Before and after the peak occurs the vacuum may be lowered and 10 degrees higher temperature than the weather requirements demand may be carried on the water system. By this arrangement starting up a second machine is avoided and a great waste of steam due to high load and low vacuum is prevented.

There are cases where it has been recommended to take steam for heating and other purposes from the receiver between the cylinders of a reciprocating engine and from a turbine between the stages. This has resulted in a number of fallacies as to economy. The governor on a large engine will not regulate for a greater range in total steam consumption than the equivalent of no vacuum and full vacuum, or from 25 to 30 per cent. and generally less. When the pressure on the receiver of a reciprocating engine is lowered sufficiently to require a reducing-valve connection, the steam may be taken from the main steam pipe with as good results. The turbine will likewise show an increase in steam consumption per kilowatt when the stages are tapped that will indicate a reduced economy from the practice. It can be done, however, if less than 10 or 15 per cent. of the steam required to operate the machine is taken in this manner. If the turbine or engine is especially designed with high-pressure parts of greater capacity for this work the machine will be uneconomical if at any time the conditions are reversed and the steam is not needed in the heating system. In any case it is the consensus of opinion that taking the steam in the manner indicated between the engine and condenser is far more economical than tapping the stages of the turbine or receiver of the engine.

Hot-water heating would be especially advantageous in connection with low-pressure turbines and reciprocating engines, using cooling towers in case injection water was not available. The charts show that over 50 per cent. of the exhaust steam required in zero weather for heating would be available for power 80 per cent. of the time.

The writer is indebted to W. E. Van Patten, chief engineer at the Hoboken terminal, for the records of operation.

TABLE 3. COST OF HEATING

Areas of Fig. 4	Pounds Steam	Per Cent.	Cost at 17c. per M.
T, heating and power, hot water	18,549,400	100	$3,153
H, heating, hot water	15,364,260	83	2,612
T', turbine at 400 kilowatts, full vacuum	12,730,800	68	2,165
$T-T'$, cost of heating by turbine, partial vacuum	5,818,600	32	989
$T-H$, excess steam for extension of work on condenser	3,185,140	17.2	741
T'', turbine, 400 kilowatts load, atmosphere	27,810,000	150	4,588
H', heating 1545 hours, medium, 210 degrees	26,265,000	142	4,465
$T''-T'$, cost of heating by turbine at atmosphere	15,079,200		2,563
$R'-T'$, economy of turbine over reciprocating engine, summer	1,328,700	7	226
$T'+H$, heating by live steam, hot water and condensing turbine	28,095,060	151	4,776
Areas of Fig. 5			
H, hot-water heating, 3,491 hours	36,102,377	100	6,140
H', heating, medium 210 degrees, 3,491 hours	59,347,000	164	10,090

First Annual Meeting of the Institute of Operating Engineers

The Institute of Operating Engineers, which was incorporated under the laws of the State of New York on June 22, has held its first annual meeting. The first session was opened with a brief speech by Prof. W. D. Ennis at 10 a.m., Friday, September 1, in the Engineering Societies building, West Thirty-ninth street, New York City. Some 50 delegates and members were present.

Ex-Congressman William S. Bennet gave the address of welcome. In speaking of the Institute he said that it typified the old American spirit of progress and he felt that there was a bright and useful future for it.

J. C. Jurgensen, founder of the Institute and provisional chairman, made the reply. He went a little into history, telling how the idea of the Institute was developed.

Several short addresses followed Mr. Jurgensen's speech.

Fred R. Low spoke on "The Operating Engineer's Future." Mr. Low said that one great trouble of the present day is to emphasize the difference between the man who simply runs his engine and the man who brings to bear upon the operation of his plant all the training, thought, interest and experience of a real engineer. To do this is one of the cardinal purposes of the Institute.

The increasing opportunities of the thoroughly trained power-plant engineer make it more than ever worth while to study and train. The increasing importance, responsibility and remuneration of such a man will aid in putting the whole vocation on a higher plane.

D. B. Heilman, of Reading, Penn., chairman and commissioner of District No. 3, said a few words on "The Engineer's Place in the Community." He told how the general public seldom gave thought to the man who bears the responsibility of keeping the wheels turning. This state of affairs is passing, however, because of the wider dissemination of information and because of the increasing uses and importance of mechanical power. The Institute will serve to hasten the time when the engineer will be accepted at his true value and his importance in the community will be fully realized and acknowledged.

A. C. Dougall was to have spoken on "The Employer and the Engineer," but was prevented from attending the meeting. A. R. Maujer spoke extemporaneously in his stead. He said that there seemed to be a lack of proper sympathy between the employer and the engineer. This was due, he thought, to ignorance; ignorance on the part of both. The blame rests, perhaps, chiefly with the engineer; first, in not taking full advantage of his opportunities, and, second, in not demonstrating his worth and importance to his employer.

J. P. Fleming, of Chicago, chairman of the T. J. Waters Branch, spoke briefly of the progress which has been made by the Institute in Chicago. The Waters Branch expects to offer a course of high-grade lectures on steam-engineering subjects during the coming fall and winter. The outlook seems good for increased membership in the Chicago district and valuable activity is expected in the way of educational work.

After the speeches the following committees were appointed: Committee on constitution and bylaws, A. L. Rice, Chicago; W. G. Freer, New York; J. A. Pratt, Philadelphia; Willis Lawrence, New York, and V. L. Rupp, Philadelphia. Nominating committee: R. D. Tomlinson, Milwaukee; J. L. MacVicar, Boston; H. M. Elder, Richmond, Va., and Fred R. Low, New York. Finance committee: D. B. Heilman, Reading, Penn.; J. C. Stewart, New York; R. D. Tomlinson, Milwaukee; J. P. Fleming, Chicago, and H. W. Geare, New York.

At the afternoon session three technical papers were read. The first one, "Temperature Changes and Heat Transmission," by V. L. Rupp, will be found elsewhere in this issue. The second, "Boiler Room Analysis of Coal," by J. P. Fleming, consisted of a description of a simple method of making a proximate analysis with a fair degree of accuracy without employing any very expensive pieces of apparatus. The main point seemed to be the use of large samples so as to offset errors in weighing. The third paper, "Cooling Towers versus Steam Pumps," by H. W. Geare, although afflicted with an irrational title, proved to be interesting, because it illustrated what a little study of conditions and plant layout may produce in the way of improved economy when applied. The paper described how changing from the use of sea water for condensing purposes to fresh water recooled in forced-draft cooling towers effected a direct saving of nearly $10 per 24 hours besides producing more satisfactory and reliable operating conditions.

In the original installation the sea water was pumped about 1000 feet to the condensers by duplex steam pumps. The friction and radiation losses and the high steam consumption of the pumps resulted in very poor economy. By installing two forced-draft cooling towers of such ample size that for eight months in the year they may be used as natural-draft towers and by using electrically driven circulating pumps much better economy was secured.

The discussion of these papers was brief, principally on account of their nature, and an early adjournment of the session was made in order to allow the various committees to get to their work.

At the evening session Professor Ennis gave an able address on "The Professional Spirit," and J. A. Pratt, director of the Williamson Trade School, delivered a lecture on "A Method of Teaching Operating Engineering." Professor Ennis' address will be printed complete in a later issue. Mr. Pratt's lecture, which was illustrated with stereopticon views, described the methods employed in teaching operating engineering at the Williamson school. The preface to the lecture proper was a rather complete survey of the present status of operating engineering and an expression of Mr. Pratt's opinion of the value of the Institute in raising both the vocation and the men to a higher plane. An abstract of his lecture will be published later.

The meeting of Saturday morning was devoted entirely to the transaction of business. Messrs. Heyrodt, Briner and Sword were appointed ballot tellers and instructed to retire to examine and report on the postal vote recently cast by the Institute membership for the election of the candidates to the various grades. They reported that all members listed on the ballot were elected to the grades under which they had been scheduled.

The treasurer's report was then made. From March 10, 1910, to September 1, this year, the total receipts for initiation fees and dues were $2284.41. The total expenses were $2156.65. In the course of his report, the treasurer pointed out that the expenses during the past year were necessarily high; office furniture, a typewriter and a duplicating machine were purchased, the prospectus pamphlets were printed and much postage expense was incurred through the dissemination of missionary literature.

The report of the committee on educational requirements was next heard. This committee has compiled and had printed a list of requirements for the various grades, which is to be used as a guide in grading the candidates. A copy of the list may be obtained by application to the secretary.

The committee on apprenticeship requirements has prepared a similar list which also is obtainable from the secretary.

The committee on vocational statistics was compelled to suspend its activities because of greater need for the efforts of its members in other directions. It hopes, however, to get local committees started in the near future for the collecting of data on the size and condition of plants, wages, hours of work, etc. During the past year this committee has been called upon to place 17 men. In one case an Institute member was sent to a plant in New Hampshire to take a position where he is now drawing a salary of $240 per month.

The finance committee reported that the books of the treasurer had been examined and found in good order.

The committee on constitution and bylaws brought in a report in which many changes in the constitution were recommended. The governing idea of the committee was to simplify the constitution as much as possible and transfer to the bylaws all matter which is not strictly fundamental law.

After a long discussion, in which strong arguments were advanced by both sides, a motion was sustained changing from 14 to 12 the number of years of practical experience required of a candidate for the degree of master operating engineer. Provision was made to make a completed four-year course in a recognized college equivalent to two years of practical experience. Another proposed change gives the power of election of all candidates for all grades to the board of control. Heretofore candidates have been elected by the vote of the entire membership. The initiation fee for journeyman machinery operators was reduced from $10 to $5; for the senior apprentices, from $5 to $3 and the registration fee for the junior apprentices was reduced from $5 to $2. The quarterly dues of the machinery operators were reduced from $2.50 to $1.50 and of the senior apprentices from $1.50 to $1.

The nominating committee submitted the following list of nominees for office and they were unanimously elected: President, J. C. Jurgensen; vice-president, for one year, Willis Lawrence; vice-president, for two years, W. D. Ennis; vice-president, for three years, J. G. Ould; secretary, H. E. Collins, and treasurer, W. P. F. Hill.

It will be noticed that all officers of the temporary organization were reëlected. In recommending the reëlection the nominating committee pointed to the great amount of work performed by these men and the progress made under their regimé.

All of the papers read at the Saturday afternoon session brought forth much discussion, for the reason, perhaps, that the topics were relative to a phase of steam engineering which is naturally open to much discussion, to wit, lubrication.

R. D. Tomlinson's paper, "Engine Lubrication," found elsewhere in this issue, engendered a discussion as to the manner of applying cylinder oil. F. L. Johnson maintained that atomization of the oil was not necessary, stating that it was the valve parts and cylinder walls that required lubricating and not the steam. He said that where the oil is atomized, 90 per cent. or more is carried in the steam out through the exhaust pipe without ever coming in contact with the rubbing surfaces at all. F. J. Curry supported the atomization method and described an experience to bear him out. In a certain vertical cross-compound engine the oil was applied by direct injection close to the throttle valve, passing into the valve chest in solid drops. The engine valve was slightly loose and this permitted the oil to leak through the valve, trickle down the valve stem and out through the stuffing box, never reaching the cylinder at all. Naturally, trouble resulted. The oil-feed pipe was tapped into the steam pipe higher up and carried in to about the center. Thus the drops of oil fell out of the feed pipe into the center and hottest part of the stream of steam, became vaporized and were swept into the cylinder along with the steam itself. The trouble ceased after the change.

W. G. Freer, who has had long experience as a marine engineer, advocated no oil whatever in steam-engine cylinders. He confessed, however, to using cylinder oil in the high-speed horizontal generator engines of which he now has charge.

H. M. Elder testified that he had found grease to be more economical than oil in some uses.

The second paper, prepared by G. L. Fales and read by F. L. Johnson, was entitled "Reduction of Lubricating Costs in Smelter Power Plants." The paper described how an impressive monetary saving was effected by the substitution of a flooding system of oiling for sets of individual sight-feed cups. The flooding system described in Mr. Fales' paper was of the pressure type; that is, the oil was fed to the parts under an air pressure of some 15 pounds. Willis Lawrence stated that while the pressure system may have been satisfactory in the present instance, the gravity system was much to be preferred in the average case. With the pressure system trouble is often experienced with the pump so churning the oil in injecting it into the pressure tank that it does not clear itself readily of the air and is fed to the parts in a more or less emulsified condition.

Mr. Tomlinson pointed out that with a gravity system where the storage tank is of fair capacity, the oil supply does not depend solely on the continuous operation of a single pump. If the pump breaks down a little time is available to get another pump into service or even to make repairs on the broken one.

An interesting bit of irrelevant discussion was started by Mr. Lawrence. He told how occasionally on large shafts a patch, sometimes as large as a man's hand, would appear, seemingly so smooth that the lubricating oil would not adhere to it. As the shaft revolved this patch would come in contact with places in the bearing which were poorly lubricated, stick and cause heating. The surest and best cure for this trouble, Mr. Lawrence found, was to rub the bright spot with a cake of Sapolio. This would roughen the spot and restore satisfactory lubrication. Mr. Johnson's remedy for a hot bearing which had developed in the manner described by Mr. Lawrence was to draw-file the shaft in a direction parallel to its length. Mr. Lawrence allowed that this was good, but argued that it was not always feasible as the trouble might develop when the engine could not be shut down or the engine might be of such size that the lifting of the bearing caps would be an undertaking for several men and a sizable crane.

The last paper, prepared by Darrow Sage and read by H. E. Collins, was on "Removing Emulsified Oil from Condensed Water." It is printed in this issue.

By way of starting the discussion, Mr. Lawrence asked: "Is it necessary to remove oil from boiler-feed water? If so, how much? In other words, what percentage of oil may safely be admitted with the feed water?" His opinion is that even a large quantity of oil is not dangerous. He told of operating a water-tube boiler at 300 per cent. rating and under 200 pounds pressure with ¼ inch of oil on the tubes. In opposing Mr. Lawrence's contention that oil was not dangerous, Mr. MacVicar told of a tubular boiler in which the lower front sheet bagged down due to but a slight film of oil on the metal. Mr. Low said that oil mixed with scale-forming feed water unquestionably was a source of trouble, the scale and oil in a physical combination forming a plastic mass which when adhering to the boiler shell excluded the water so as to permit the metal to become overheated

On Saturday evening practically all of the out-of-town members and a large number of the resident members visited the Fifty-ninth street station of the Interborough Rapid Transit Company. This is the station in which the first notable installation of exhaust-steam turbines was made. Mr. Lawrence is the chief engineer for the Interborough company.

On Sunday the out-of-town members were piloted about the city by the reception committee to various points of interest.

All who attended this first annual meeting were impressed by the headway which the Institute has thus far made and expressed the fullest confidence that even greater strides will be made during the coming year.

PERSONAL

John S. Baker, well known among the electrical trade on the Pacific coast, has been placed in charge of the offices in San Francisco, Cal., Room 400, First National Bank building, opened on September 1 by the Crocker-Wheeler Company, of Ampere, N. J. Motors, generators and transformers of various sizes will be carried in stock for coast shipments.

Lieutenant Walter Diman, son of George H. Diman, consulting engineer for the American Woolen Company, has been appointed senior engineer of the battleship "New Jersey."

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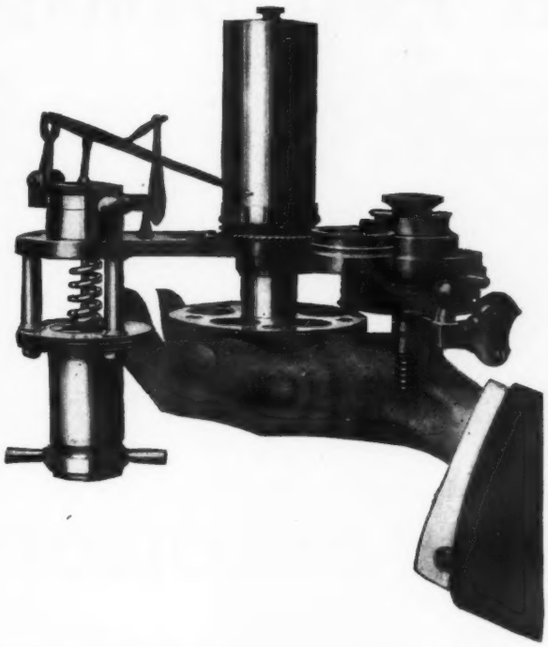

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Look over the list of our customers which we have given you the last two or three weeks. The requirements of these firms in the indicator line are the very highest. Not a **"penny wise or pound foolish"** name among them.

At the same time, mark you, they were not above the saving of $15.00 or $20.00 on each outfit, what it cost to build the instruments in a large city, over what it does in a small city in our own factory building, with natural gas for power at less than 50c. per horse power.

If you want the highest grade indicator, use the Triumph. If you want the most value per dollar in your indicator, buy the Triumph. Anyway, get the Triumph on 10 days' approval, to be returned at our expense if you do not find it satisfactory, and then some.

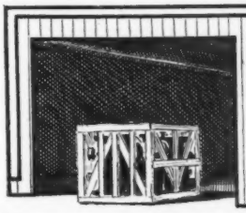
Trill Indicator Co.,
Corry, Pa.

Trill Indicator Co., Corry, Pa,

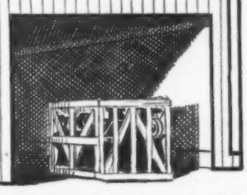
Please send me the full story of Trill Triumph Indicators, together with a copy of your guarantee and selling plan.

Name.....

Address.....



Second-Hand Bargains



We offer Electrical and Steam Machinery, of Standard makes, guaranteed as to condition. Personal inspection and investigation are invited. We can fill practically any requirement. Tell us your needs today.

50% SAVED

SPECIAL BARGAINS, IMMEDIATE DELIVERY

1—1750 K. W., 60 cy., 3 ph., 2300 volt Alternator, D. C. to Cross Comp. H. D.
Allis-Chalmers Corliss Engine.
1—300 H. P. B. & W. Water Tube Boiler, 150 pounds.
1—380 H. P. B. & W. Steel Header with Stoker.
1—500 H. P. Sterling High Pressure with Stoker.
2—515 H. P. B. & W. Steel Header with Stoker.

The Paul Stewart Company, First National Bank Building, **Cincinnati, Ohio**

417 PINE ST. Second Hand Apparatus of Any Kind **ST. LOUIS**

ARTHUR S. PARTRIDGE

BOILERS

1—350 H.P. HEINE—140 lbs. Pressure
2—250 H.P. HEINE—150 lbs. Pressure
4—200 H.P. HEINE—160 lbs. Pressure
4—225 H.P. B. & W.—135 lbs. Pressure

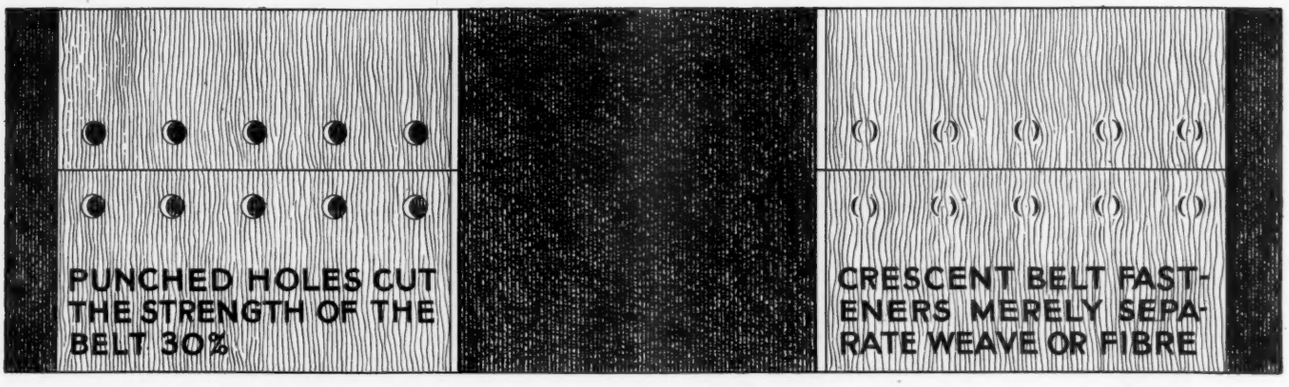
CORLISS ENGINES

1400 H.P. Hamilton Cross Comp. Heavy Duty, Cylinders, 26″ x 42″ x 54″. Wheel, 17′ diam.
750 H.P. Filer & Stowell Cross Comp. Heavy Duty, Cylinders, 22″ x 38″ x 48″. Belt Wheel 20′ x 60″.

A New Department

This is a new department. If you have any near new or second hand power plant apparatus to sell—this is the place to advertise it.

When A Belt Is Punched For Lacing It Has Received Its First Damage



Crescent Belt Fasteners

"Holds Without Holes"

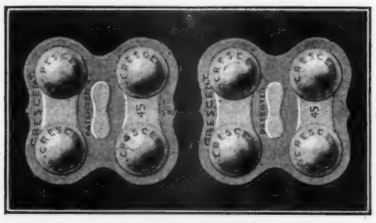
In the use of lacing from 12% to 30% of belt width is removed with a corresponding loss of belt strength. A finished laced joint is at the weakest part of the belt.

The prongs of Crescent Rivets merely separate the lengthwise fibres, thereby maintaining the maximum strength of the belt. A finished Crescent joint is as strong as any other part of the belt.

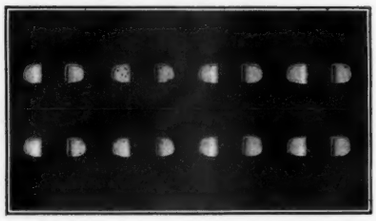
Of course it appeals to your sense—because you see at a glance that it's the logical way to join a belt.

Thousands of factories are using Crescent Belt Fasteners to-day because some one saw their many advantages, and having given them a trial was convinced that in the majority of cases they are as good as an endless belt.

Outside of Belt



Pulley Side of Belt



The only labor is to square the ends of the belt and drive Crescent Rivets, which are self-piercing, through the plate and belt and clinch with an ordinary hammer. The prongs clinch below the surface of the belt, thereby giving the belt an uninterrupted grip on the pulley.

To shorten a belt necessary through slack simply remove the rivets from one end of plate, cut out the slack and re-join at the small cost of a row of Crescent Rivets. This method insures a one joint belt.

You pay your good money for a good belt; why not insure its satisfactory service by using Crescent Belt Fasteners which have gained the endorsement of and are recommended by the principal belting manufacturers?

Adopt the service that Crescent Belt Fasteners alone can give.

Every Crescent Plate is stamped "Crescent" and the head of every Crescent Rivet is marked "Crescent." See that you get the genuine.

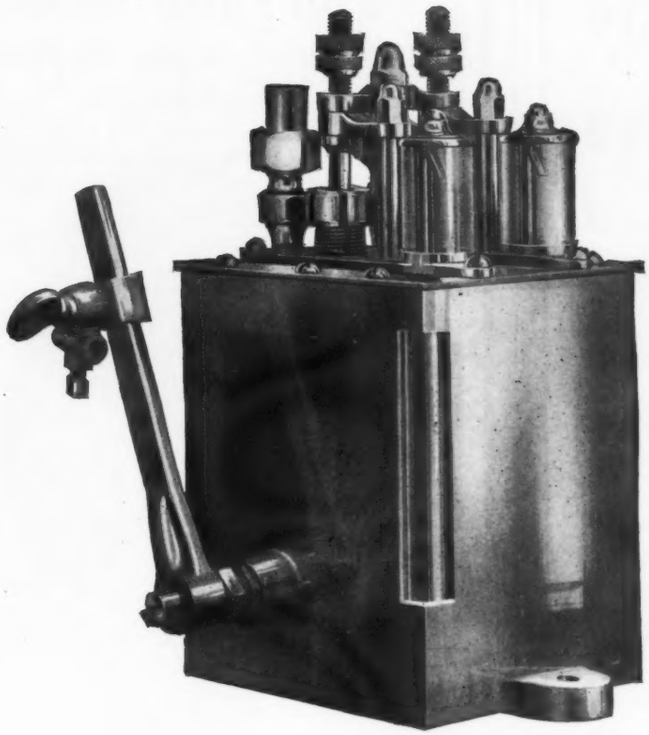
Send For Free Sample

Advise us the width and thickness of a belt on which you will try Crescents and we will send the proper fastener without cost to you. Mention the diameters of the pulleys.

Crescent Belt Fastener Co.,

381-385 Fourth Avenue
New York City.

Patentees and Sole Manufacturers



"McCORD" FORCE FEED LUBRICATORS

INCREASE EFFICIENCY DECREASE OIL EXPENDITURES

McCord Lubricators insure a positive delivery of clean, fresh oil to the bearings. Any other system of lubrication involves a re-circulation of the same oil. This re-use of oil is not conducive to the best results in the lubrication of frictional parts because heat, pressure and the continual churning impair its value as a lubricant.

The feeding of oil with enough force to overcome the pressure developed in steam chests, cylinders, etc., is the main result to be obtained, yet a device which cannot be quickly and easily adjusted to conform with the varied requirements of the engine and which does not have a positive and simple sight feed, is worse than useless.

McCord Force Feed Lubricators embody all these essentials and many other desirable features: all of which combine to make it the best and best-known Lubricator today.

McCord Force Feed Lubricators Give High-Grade Service at a Reasonable Price.

If you value your engine, give it McCord Force Feed Lubrication.

Catalog "B.L." is published for your benefit.

McCord Manufacturing Company

New York **Detroit, Michigan** **Chicago**

AGENCIES:

Boston Pittsburgh Richmond St. Louis San Francisco Seattle Denver St. Paul

In These Rapidly Shortening Days

when you need more and more power and light at the same time

¶ When, with the earlier sundown, it becomes more and more necessary to light up the building often long before the machines are stopped, always just at the time when the demand for rapid elevator service is heaviest—

¶ The full value of The Edison Service to the engineers in charge of the plant is increasingly apparent.

¶ For with The Edison Service, the engineer is able to meet these early morning and late afternoon demands of the autumn and winter fully and economically.

¶ A turn of a switch and there is plenty of power just when power is needed; and this power is secured without the necessity of having a large equipment lying idle—and eating up money—during those other hours of the day when the load is bound to be less.

¶ And for you, the Engineer, to introduce and maintain a service of this kind, is to make yourself a more valuable man to your employer.

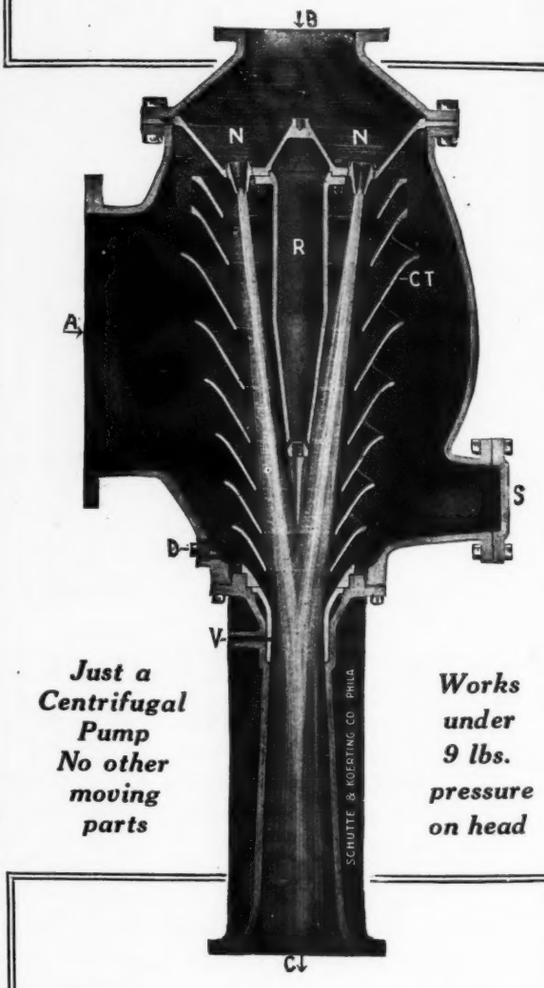
¶ *You can get full details by writing us—or 'phoning Worth* 3000.

The New York Edison Co

55 Duane Street New York City

No Air Pump Necessary

With Koerting Multi-Jet Condensers



Just a Centrifugal Pump No other moving parts

Works under 9 lbs. pressure on head

FAR superior to any other condenser for use in plants where large units are being operated and high vacuum is required with minimum water consumption.

Besides obtaining and maintaining the vacuum without the use of air pumps the Koerting Multi-Jet Condenser has these superior advantages:

Maximum contact of exhaust steam and water; constant steam velocity reducing drop of vacuum between the interior of the tube and the exhaust chamber to the minimum; low consumption of power; minimum consumption of water; no moving parts; simplicity of operation; takes up little space; no wear and tear.

A highly efficient device in every sense of the word. You will be interested in the full description of design, construction and operation.

Send today for Catalog 5 *B.*

Schutte & Koerting Co.

Tompson and 12th Sts., Philadelphia

New York, 1617 Fulton Bldg.
Boston, 96 High St.
San Francisco, O. C. Goeriz & Co.
Seattle, E. P. Jamison & Co.
Chicago, 803 Security Bldg.
Pittsburg, 1718 Keenan Bldg.
Cleveland, New England Bldg.
Denver, 1710 Glenarm St.
Portland, E. P. Jamison & Co.

In the interest of *safety* alone, make sure that specifications for your steel valves and fittings read thus:

"The castings must be **NELSON** Steel."

Because **NELSON** Steel is the *only* steel made that is *guaranteed* for valves and fittings for Superheated Steam.

In the **NELSON** Steel Foundry no Castings are made except for pressure work. The sole requirement is *quality*, no matter what the quantity or cost. *Other Steels* are made in foundries where they make anything from locomotive frames to car couplers, where only large quantities of *cheap* castings is the main purpose. The grade is low, because the castings are cheap.

Tapping a NELSON Steel Furnace

We are the *only* Valve Company that has *its own* open hearth steel foundry. We were driven to make our own steel because we couldn't buy guaranteed steel castings from any steel foundry in the United States. We found that, with skill and an extra thorough carefulness, steel castings *could* be made, and made *regularly*, back of which we could safely put the **NELSON** guarantee.

This is one strong reason for the general preference for the **NELSON** Steel Valves.

But if you decide to use Steel Valves or Fittings other than **NELSON**, *remember* that we are willing to make the **NELSON** Steel for other Valve Companies if you will *specify* it.

Nelson Valve Company

Philadelphia

NEW YORK 30 Church St.
CHICAGO 17 West Kinzie St.
SAN FRANCISCO 22 Battery St.
PITTSBURG 517 Liberty Ave.
MINNEAPOLIS 622 Andrus Bldg.
BUTTE 56 E. Broadway
MONTREAL 12-14 University St.

A High Score—Always

PIPE COILS & BENDS

Of any shape, size or style and of iron, steel, brass, or copper. All work of the highest quality.

All work done on patented electrical and hydraulic machinery without wrinkling, flattening, or distorting the pipe. Our facilities are the largest and most up-to-date of their kind. Our experience is large and varied and will be of great value to you. *Write for catalog of more common coils and bends together with full information.*

The Whitlock Coil Pipe Co., Hartford, Conn.

Main Office, Hartford, Conn. New York Office, Singer Bldg.

Forbes Motor Driven Pipe Threading and Cutting Machine

Reduces cost of pipe cleaning and cutting Saves power.

Power can be shut off when machine is not in use. Motor is concealed in cabinet case.

Machine can be taken from place to place. Easy to operate, accurate, it is an expense saver of the highest order.

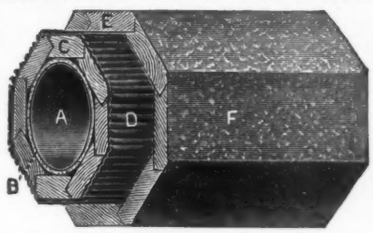
Write for catalog

The Curtis & Curtis Co.
85 Garden St.,
Bridgeport, Conn., U.S.A.

No Radiation Of Heat

There can be no radiation of heat—no loss of steam—where

Wyckoff Patent Water Proof Steam Pipe Covering



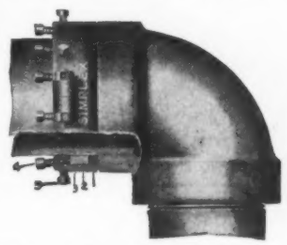
is used because this covering is made of materials which are absolutely non-conductive and which make a perfect pipe possible. *Write for the Wyckoff Red Book.*

A. Wyckoff & Son Co., **Elmira, N. Y.**

Sure Cure For Leaks

Force a square packing against the leaky joint. Then force a ring against the packing. Lastly bolt an outside ring around the first ring. Secure to pipe with set screw and your **Simplex Pipe Clamp** is adjusted. It is the easiest, quickest, cheapest and most permanent way of stopping leaks. Simplex Clamps are sent on 30 days' trial.

Write for Clamp Catalog.

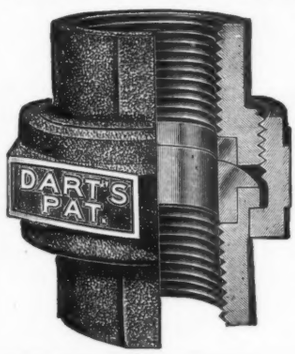


Yarnall-Waring Co., 1111 Locust St., Philadelphia, Pa.

Formerly The Simplex Engineering Co. Name changed because of confusion among various "Simplex" companies.

A Sample Will Prove **"Bronze to Bronze"**

Means No Corrosion



Let us send you a sample Dart Union, free. Examine it carefully. Note the extra strength of the pipe ends and nut. And *consider* carefully the advantages of *two* bronze seats compared to one or none at all.

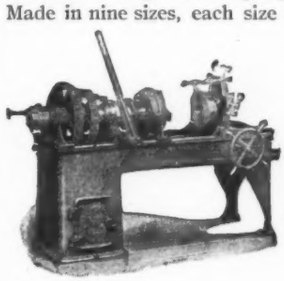
We have a new booklet to send you showing the different styles of Dart Unions with prices. *Write for it today.*

E. M. Dart M'f'g Co.
Providence, R. I.

Canadian Factory—Dart Union Co., Ltd., Toronto.

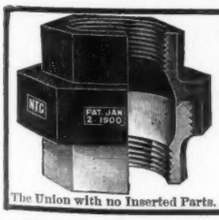
Clean, Accurate Pipe Threads

are the certain result if you use a B. & K Pipe Threading Machine. They work rapidly and are easy to operate too. Made in nine sizes, each size having a capacity of from 8 to 10 sizes of pipe. Our machines are equipped with the Peerless Die Head.



Send for catalog.

Bignall & Keeler Manufacturing Co.
Edwardsville, Ill.



The Union with no Inserted Parts.

Every "Kewanee" Union Is Tested

Tested before shipment to 125 lbs. compressed air under water. Ask for *"The Whole Kewanee Family"* booklet. It's free!

National Tube Company,
Pittsburgh, Pa.

Unlike Globe And Gate Valves

The Homestead Valve Seat is absolutely protected from wear as the pressure passes through it.

Homestead Valve Mfg. Co.,
Brass Founders **Pittsburg, Pa**

New Valve Catalog Now Ready For Distribution

Write for a copy

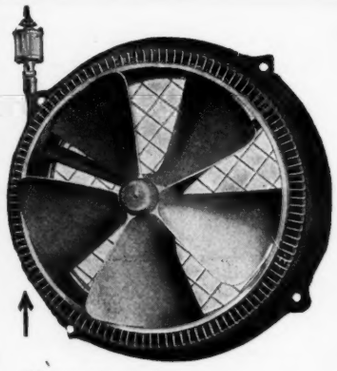
Chapman Valve Mfg. Company,
Indian Orchard, Mass.

Ever have broken stems on your valves?

Well, you'll never be troubled with them again if you will install—
THE MONARCH LOOSE NECK GATE VALVE.
For the loose neck feature absolutely guards against broken stems. The *proof* is that we've *never* had a valve returned to us with a broken stem.

Post card brings full details.

New York Boston **Monarch Valve & Mfg. Co.** Springfield, Mass.



Where power is applied Casing Removed

Why pay for coal that you don't burn?

If you are not securing complete combustion of your fuel you are simply spending a lot of money for which you get no return.

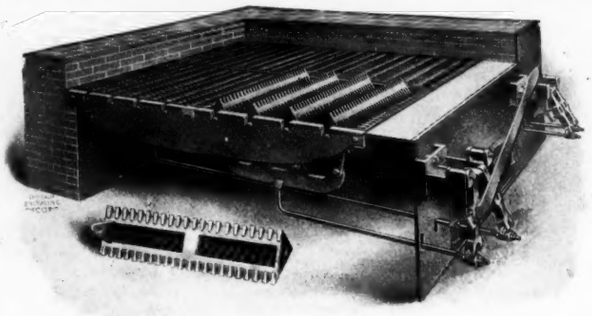
The "Typhoon" system of Mechanical Draft makes coal waste due to incomplete combustion impossible.

Assures steady steam pressure. INCREASES BOILER CAPACITY. PERMITS BURNING CHEAPER GRADES OF FUELS ECONOMICALLY. The big companies use it because it is a money saver; the smaller companies can use it with equal results. Bulletin No. 7 tells all about it. Want a copy?

L. J. Wing Mfg. Co., 90 West Street, New York

BOSTON PHILADELPHIA CHICAGO

A Feature That Is Valuable To You And To Us



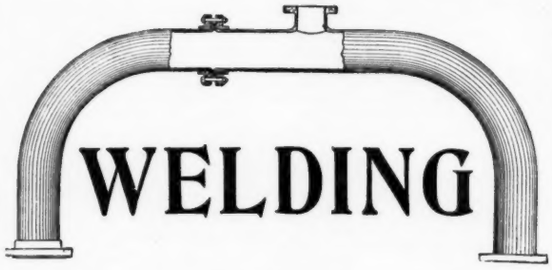
2 Grates in 1

Every bar in the Long Duplex Shaking Grate has *two usable* sides, one in use while the other is below fire cooling off. In this way each side of the bar is heated in like degree, preventing warping, and, it has been proven, making the grate fully as long lasting as two ordinary grates.

There's value there for you because you pay for one grate and practically get two. There's value for us because by being able to give our customers two grates in one at the price of one we increase our sales.

Find out about this and other features. Write today.

The Long Grate Bar Company
Buffalo, N. Y.



WELDING

Makes For Greater Power-Plant Economy and Efficiency

The fewer pipe joints in your plant, the less trouble you'll have, the less expense.

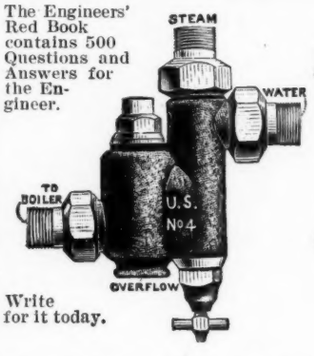
By welding outlets on piping and welding pipes, end to end, fully 33⅓% of the joints ordinarily required can be done away with.

Less attention will be needed, less pipe covering—you'll have no leaks to worry you, you'll save in many ways.

Write for our catalog No. 27.

The M. W. Kellogg Company
50 Church Street, New York

The Engineers' Red Book contains 500 Questions and Answers for the Engineer.



Write for it today.

United States Automatic Injectors

are dependable boiler feeders.

If you want to be absolutely sure that hot water will be put in your boiler when you want it, then use

U. S. AUTOMATIC INJECTORS.

We guarantee every Injector that leaves our factory.

Get our Catalog "G."

American Injector Co., Detroit, Mich.

AIR SPACE TO SUIT YOUR NEEDS

The Perfection Grate provides any air spaces you need. It is simple, strong and lasting. Cleans fires without loss of coal.

DETAILS?

Perfection Grate Co., WHITNEY B'LD'G. Springfield, Mass.

Rolin Patent Adjustable Grate Bars

Can be adjusted to any air space.
⅛"-¼"-⅜"-½"-⅝"-¾"-⅞"-1".
An air space for every grade of coal.
30 days' trial allowed on each grate.
Sold as low as $25.00 per boiler.

Standard Grate Company, 1215 Filbert St., Phila, Pa.

KEYSTONE GREASE

KEYSTONE TRADE MARK REGISTERED U. S. PAT. OFF.

You Better Take Up Our Free Offer Too

And Send at once for the Can of Keystone Grease, Grease Cup and Engineer's Cap

Because there are some things it will pay you to learn about Keystone Grease.

For Keystone Grease is all pure lubricant—composed of petroleum oils that are natural friction-reducers, scientifically put together.

Keystone Grease is insoluble in water, it contains absolutely nothing that is injurious to bearings, it has the same consistency winter and summer.

Keystone Grease is applied to bearings by pressure grease cups. The development of frictional heat is not required to cause the grease to feed. The original density of Keystone Grease never changes under the severest conditions of speed and pressure and atmospheric heat and cold. Keystone Grease holds the world's record for the lowest co-efficient of friction. No other lubricant on the market can compare with it in quality—a lubricant with thirty years' reputation back of it.

Fill out the coupon and return—now.

KEYSTONE LUBRICATING CO.

Executive Offices and Works

21st, Clearfield and Lippincott Streets **Philadelphia, Pa.**

Branch offices and warehouses in the principal cities of the United States.
Agencies in the principal countries throughout the world.

No Connection With The Oil Trust

A SAMPLE FREE

KEYSTONE LUBRICATING COMPANY, Phila.

Please send us a large size sample of Keystone Grease, sufficient for testing purposes, and one Brass Grease Cup and Free Engineer's Cap. It is thoroughly understood that no charges whatsoever will be made for sample, cup or cap, and all express charges will be paid by you.

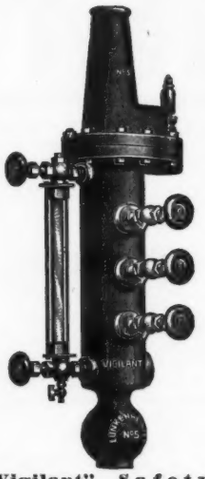
Name of bearing where sample will be tested.....
Its width.....Combined horsepower of engines.....
Tap size of cup.....
FIRM.....
Per .....
City .....State .....
Size cap required..... **CUT OUT THIS COUPON AND MAIL IT TO-DAY** Dept. B.—9-19-11

"Vigilant" Safety Water Column without water gauge or gauge cocks.

Prolong The Life Of Your Boiler

Save **Fuel Bills** and **Prevent Accidents** by installing

Lunkenheimer "Vigilant" Safety Water Columns



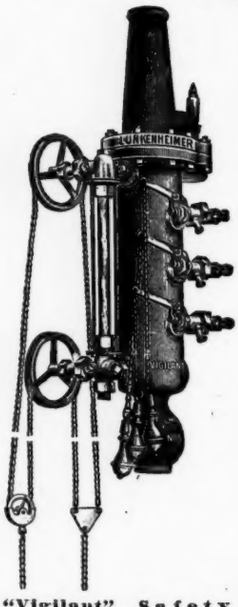
"Vigilant" Safety Water Column with regrinding gauge cocks and three rod water gauge.

They will Automatically Sound an Alarm when the water in the boiler approaches the low or high danger limit, thereby notifying the boiler attendant and Preventing Serious Accidents.

By insuring a uniform water level, the "Vigilant" effects Greater Economy in the use of fuel, Obviates Expensive Repairs and Stoppages and Lengthens the Life of the Boiler.

Only one strong, seamless copper float is used and it is guaranteed not to collapse under 350 pounds of pressure per square inch.

The "Vigilant" water columns are made of the highest grade of iron and are guaranteed for working pressures up to 250 pounds. All parts subjected to wear can readily be renewed and are made of a good bronze composition containing a high percentage of copper and tin. The columns are made in all sizes suitable for the various types of boilers.



"Vigilant" Safety Water Column with "Excelsior" gauge cocks and "Monitor" water gauge.

Write for Catalogue

"MOST supply houses sell them--yours CAN--if they DONT or WONT--tell US"

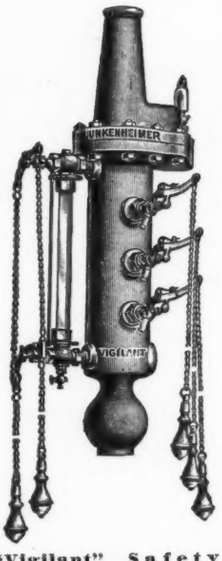
The Lunkenheimer Company

Largest Manufacturers Of High-Grade Engineering Specialties In The World

General Offices and Works

Cincinnati, Ohio, U. S. A.

New York, 64-68 Fulton St. Chicago, 186 North Dearborn St.
London, S. E., 35 Great Dover St. Boston, 138 High St.



"Vigilant" Safety Water Column with rotating gauge cocks and quick-closing water gauge.

10

Scale and Economy cannot exist in the same boilers. *Enter scale—exit economy.*

And that is true whether your plant is small or large, whether you have the most expensive boiler equipment in the world or the least expensive.

As soon as scale settles on the heating surface, efficiency decreases and coal bills increase.

In fact, you can get as much work out of a small boiler that is clean as you can out of a large boiler with scale filled tubes.

So keep the tubes clean, keep scale out of the boilers, if you want boiler economy.

And keep it out day after day, not only one day in every seven.

The only way to handle this scale problem so that you will have continually clean boilers is to treat the boiler feed water with Dearborn Treatment.

Dearborn Treatment *is not* a cure-all boiler compound.

It is made to suit the conditions that exist in your plant, after a careful analysis of your boiler feed water has shown what these conditions are.

That is why Dearborn Treatment is successful while cure-all compounds fail.

And that is why our laboratory is called upon to analyze over 8000 feed water samples annually.

Send us a gallon of your feed water and let us show you how inexpensively you can get rid of your scale troubles

Dearborn Drug & Chemical Works

Robert F. Carr, President

General Offices, Laboratories and Works
Chicago

General Eastern Offices, 299 Broadway
New York

BRANCH OFFICES IN PRINCIPAL CITIES

THE RIGHT WAY SEND FOR SMOOTH-ON INSTRUCTION BOOK FREE

It will save YOU more time and money than anything you can have in your engine or boiler room.

It shows, by photograph and formula, how to use the various well-known Smooth-On Iron Cements in repairing boilers, engines, castings, tanks, joints, etc.

The uses of SMOOTH-ON are innumerable, and its repairs absolutely permanent.

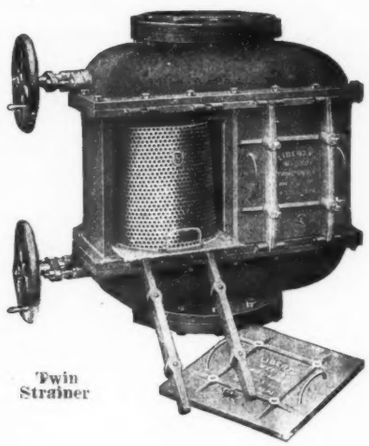
SMOOTH-ON MANUFACTURING CO.

572-574 Communipaw Ave., Jersey City, N. J.

Chicago, 231 N. Jefferson Street **San Francisco, 94 Market Street**

London, 8 White Street, Moorfields, E. C.

DO IT NOW



Twin Strainer

Don't allow another day to pass without installing a Twin Strainer.

The season is approaching when you will need one on your water line. In fact, you need one **all the time.**

There is but One Cost to the Twin, and the protection it affords you against shut-downs during one spell of high water, will equal the entire cost of the installation.

There is **no limit** to the Size.

We build Strainers to meet all conditions, and if you Don't Need One **Now** you should be posted, so you can get it quickly when you do require it.

Write for Catalog

Elliott Company

6908 Susquehanna Street, Pittsburgh, Pa.

When The Tubes Commence To Sag

It Is No Doubt Due To The Weight Of Scale

Cyclone Cleaner

Better examine your boilers **now,** and if you find the slightest trace of scale

Order a Liberty Tube Cleaner

Don't put it off until it is necessary to replace the tubes. Seventy-five per cent. of retubing is caused by scale separating the water from the metal, causing the tube to burn.

Let us tell you how to keep your tubes from sagging and burning out.

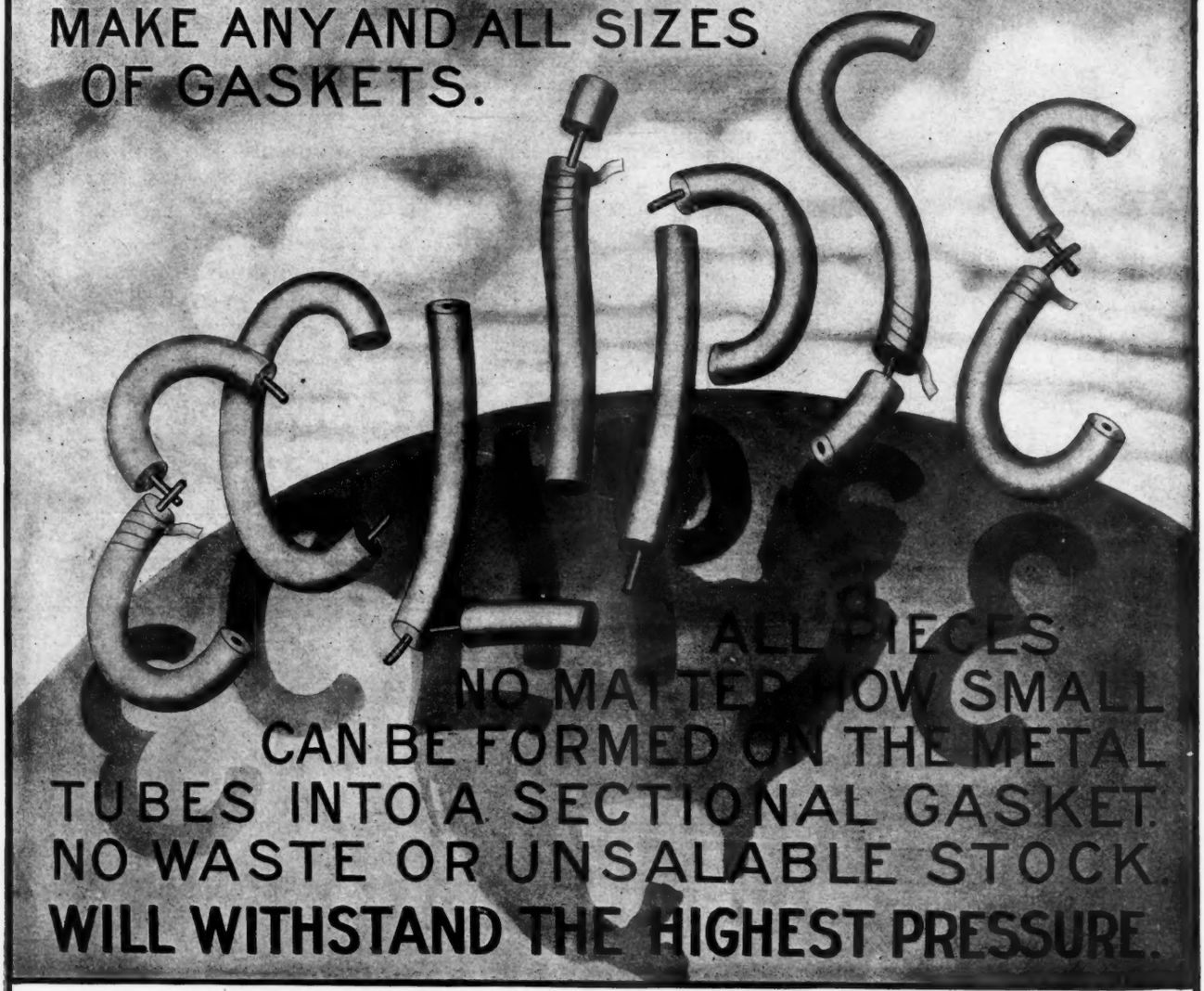
Liberty Manufacturing Co.

6705 Susquehanna Street Pittsburgh, Pa.

Bull Dog

ECLIPSE SECTIONAL RAINBOW GASKET

CAN BE ADAPTED TO ANY SHAPE AND MAKE ANY AND ALL SIZES OF GASKETS.



ALL PIECES NO MATTER HOW SMALL CAN BE FORMED ON THE METAL TUBES INTO A SECTIONAL GASKET. NO WASTE OR UNSALABLE STOCK.

WILL WITHSTAND THE HIGHEST PRESSURE.

PATENTED AND MANUFACTURED EXCLUSIVELY BY

PEERLESS RUBBER MANUFACTURING COMPANY

16 WARREN ST., NEW YORK

DETROIT, MICH:-16-24 WOODWARD AVE.
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SALT LAKE CITY, UTAH:-45-52 W. 2ND SOUTH ST.
ATLANTA, GA:-64-70 MARIETTA ST.
NEW ORLEANS, LA.,808-821 TCHOUPITOULAS COR. JULIA ST.
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PHILADELPHIA, PA:-19 NORTH SEVENTH ST.
SYRACUSE, N.Y.-212-214 SO. CLINTON ST.
ROCHESTER, N.Y.-24 EXCHANGE ST.
ST. LOUIS, MO.-454 PIERCE B'L'D'G.
LOS ANGELES, CAL:- 359 NORTH MAIN ST.
BUFFALO, N.Y.:- 379-383 WASHINGTON ST.
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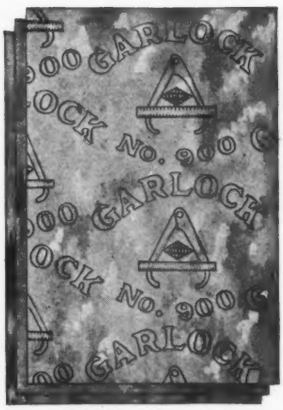
COPENHAGEN DEN:- FREDERIKSHOLMS KANAL 6
JOHANNESBURG, SOUTH AFRICA, BARSDORF B'L'D'G
PEERLESS RUBBER SELLING CO OF AUSTRALASIA, LT'D. 1 ASH STREET, SYDNEY AUSTRALIA

BECAUSE



Garlock Style No. 900 Sheet Packing for Superheat and High Pressure Steam Service.

you do not care to be furnished with arguments ONLY, we wish to demonstrate to you by

FREE TESTS

to be made in your own plant by your own mechanical men.

Garlock Red Sheet Style No. 22 for ordinary sheet packing use.

Garlock High Pressure Ring Style No. 200 for H. P. Steam Service.

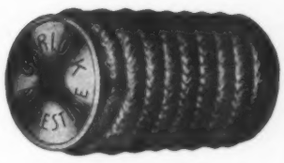
You will be convinced that Garlock Packings are the most economical that you can buy.

Garlock Diagonal Cut Ring Style No. 550 for numerous uses.

WRITE US ABOUT THIS OFFER TODAY.

Garlock W. P. H. Ring Style No. 99 for Cold Water Pistons.

Over 25 years' manufacturing experience at your disposal.



Garlock Style No. 17 Alabastine for Small H. P. Stems.

The Garlock Packing Company

Main Offices and Factories - - - PALMYRA, N. Y.

Branches in all Principal Cities.

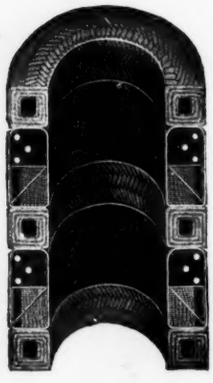
RINGS FROM OUR FAMOUS BRANDS.

STYLE 213

STYLE 102

STYLE 101

A FEW OF THOSE SHOWN IN OUR ENGINEERS' CATALOGUE. MADE TO MEET ALL CONDITIONS OF SERVICE.



STYLE 529

STYLE 528

STYLE 201

SEND FOR OUR NEW ENGINEERS' CATALOGUE

MANUFACTURED EXCLUSIVELY BY THE

NEW YORK BELTING & PACKING CO. LTD.

91-93 CHAMBERS ST. NEW YORK

CHICAGO, ILL., 130 West Lake St. ST. LOUIS, MO., 218-220 Chestnut St. PHILADELPHIA, PA., 821-823 Arch St.
SAN FRANCISCO, CAL., 129-131 First St. BOSTON, MASS., 232 Summer St. INDIANAPOLIS, IND., 120 So. Meridian St.
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LONDON, ENGLAND, 13-15 SOUTHAMPTON ROW

PENBERTHY

AUTOMATIC INJECTORS

SIMPLE

ECONOMICAL

GUARANTEED

RELIABLE

DURABLE

HIGHEST EFFICIENCY

HAS STOOD THE TEST FOR TWENTY-FIVE YEARS.

PENBERTHY

"X L-96 EJECTOR"

THE LIQUID-LIFTER

FOR conveying liquids from one receptacle or level to another, emptying fly wheel pits, sumps, flooded cellars, etc., exhausting air, gases, priming centrifugal pumps, etc., the "X L-96" *Ejector* is unequalled.

The Penberthy Safeguard Automatic Water Gage

perfectly constructed and simple in operation, gives an absolute protection against all dangers, liable to result from a breaking gage glass.

Price no more than that of an ordinary gage which lacks the desirable safeguard features.

⅜" ½"—$2.25 ¾"—$2.75

Descriptive circular matter on any of our products will be sent upon request.

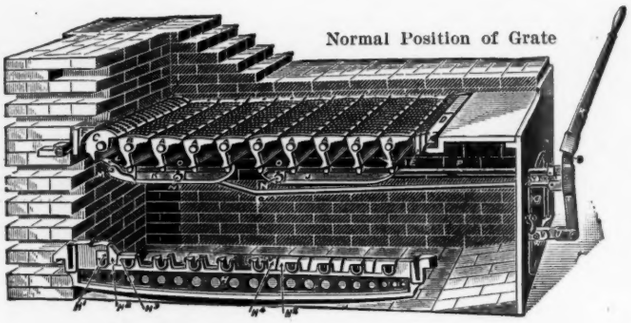
PENBERTHY

Price $2.25

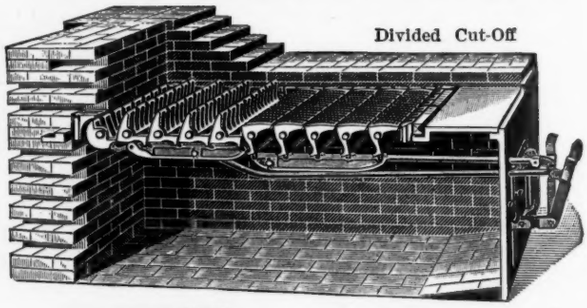
SAFEGUARD

PENBERTHY INJECTOR CO. DETROIT MICH.

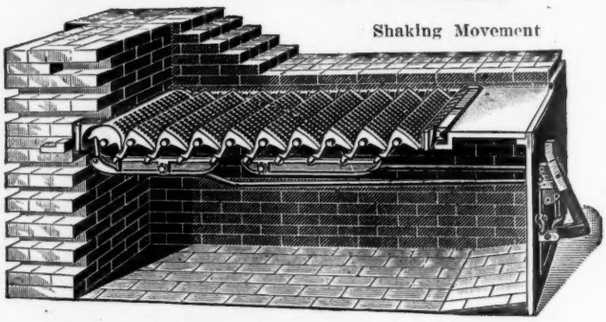
Not Only Superior Grates But A Complete System For Perfect Combustion



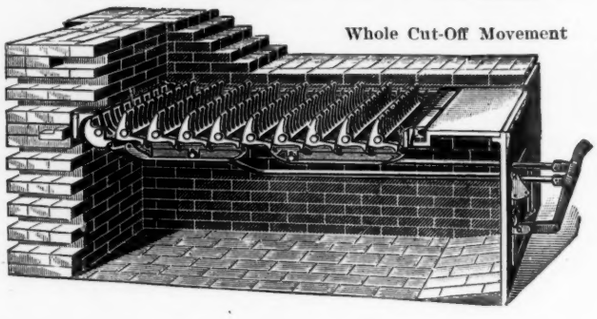
Normal Position of Grate



Divided Cut-Off



Shaking Movement



Whole Cut-Off Movement

McClave Grates with Argand Blowers

will burn successfully all kinds of cheap small fuels, such as anthracite birdseye, bituminous slack, etc.

Note the special features of the grates as shown in the cuts. The grates alone will give excellent results in burning the larger sizes of either variety of coal by natural draft.

The Argand Steam Blower furnishes a large amount of air with a small amount of steam.

The air and the steam are thoroughly mixed in the shell of the Blower before the blast is delivered into the ash-pit.

It makes very little noise in operation and furnishes a superior quality of blast for the burning of small coal fuels.

Write for full details—ask us to send Catalog "G."

Argand Steam Blower

McClave-Brooks Co., Scranton, Pa.

New York, 351 Fulton Bldg.
S. C. Smith, Manager.

Branch Offices:
Pittsburg, 1007 Empire Bldg.
Chas. N. Hays, Sales Agent.

Chicago, 706 Fisher Bldg.
F. G. Smith, Manager.

A PHILADELPHIA GREASE CUP FREE

Together With One of These Samples of Grease

Sign The Coupon

Engineers act quickly and profit by this exceptional offer. For a limited time we are offering you absolutely free one of our **"Philadelphia" Compressed Air Grease Cups,** with a free sample can of any kind of grease you desire. We will send these prepaid to you, if you will fill out and mail to us the attached coupon.

Some Reasons Why It Will Please You

In the first place we guarantee the **"Philadelphia"** Cup to save you between 25 and 50% over any other make of grease cup, if you are now using grease, and between 50 and 75% if you are now using oil.

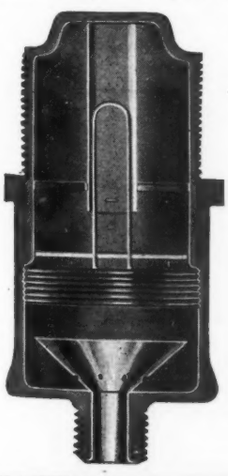
The **"Philadelphia"** is not a new grease cup. It has been on the market for several years, and has been proved and tested with hard service under all kinds of conditions, by engineers all over the country. They verify our claims.

The **"Philadelphia"** Cup is a departure from other types of grease cups on the market, and is the only automatic cup which operates by compressed air.

The advantage of this is as follows: First, it reduces time, labor and operating attention to a minimum; Second, the compressed air insures a uniform pressure from the time the cup is full until the last drop of grease is consumed; Third, the uniform pressure maintains the original consistency of the grease, which effects a saving that no other cup is able to accomplish; and Fourth, the cup is positive in action whether on a stationary bearing or moving parts where it has to overcome centrifugal forces, reciprocating action, or vibration.

Our Reasons For Making This Offer

We are anxious to give all engineers one of these cups because we want you tc make actual tests and comparisons, which will prove to you without a doubt all our claims for the **Philadelphia Compressed Air Grease Cup.** We have a cup that will positively insure the grease, in its natural state, to reach the point of lubrication, and to accomplish its full work effectively and economically. We will send you the cup and grease free to prove it.



All good cup greases will work satisfactorily in **"Philadelphia" cups.** The following manufacturers have agreed to co-operate with us in sending samples:

Philadelphia Grease, manufactured by Philadelphia Grease Mfg. Co., Philadelphia, Pa.
Zorozo Grease, manufactured by F. S. Walton Co., Philadelphia, Pa.
Albany Grease, manufactured by Adam Cook's Sons, New York, N. Y.
Lubrantine, manufactured by Wm. C. Robinson & Son Co., Baltimore, Md.
Non-Fluid Oil, manufactured by N. Y. & N. J. Lubricant Co., New York, N. Y.

But remember any other grease that you may prefer, just name in the coupon, and we'll try to supply it.

Just sign and mail us the coupon properly filled out, and get the free cup and any grease you desire to try in it.

Universal Lubricator Co.
1337 Arch Street, Philadelphia, Pa.

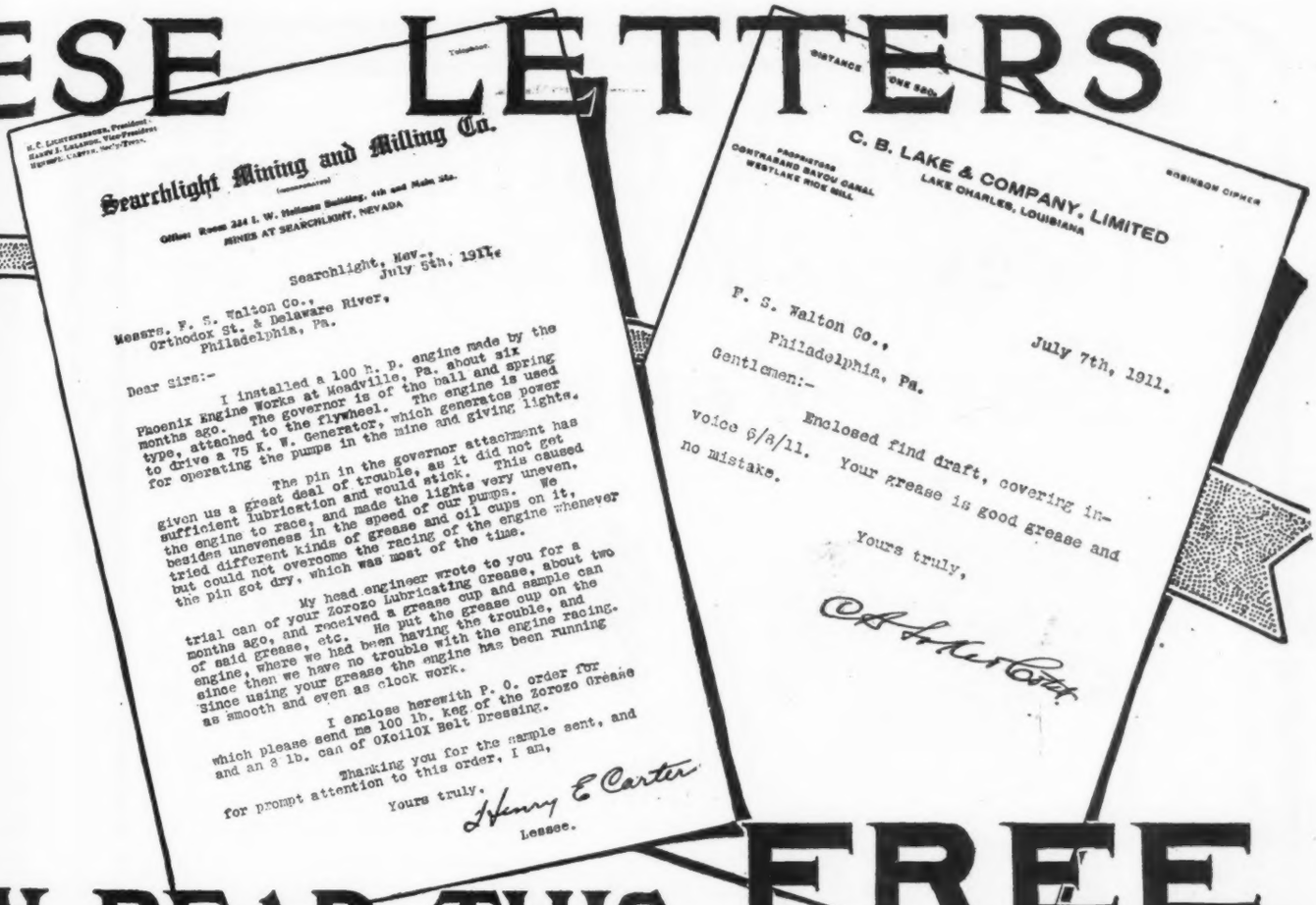
COUPON

Universal Lubricator Co., Philadelphia, Pa.
Name .....
Firm Name .....
Street Address .....
City .....
County .....
State .....
Our buyer's name is .....
State on which bearing feet will be made .....
Diameter .....
H. P. M. .....
Size of tap .....
Which grease .....
Name of dealer .....
Dealer's address .....

POWER, 9-19-11

READ TH
ZOROZO
LUBRICATING GREASE
GRADE No. 1
FOR GENERAL USE.
MANUFACTURED EXCLUSIVELY BY
E.S. WALTON COMPANY
PRESSERS AND REFINERS OF ANIMAL OILS.
THE
Here's Another
FREE OFFER
If You Prefer This One Drop Us A Line
We will send you a barrel of Zorozo and a dozen "Philadelphia" Compressed Air Grease Cups for ninety days' free trial, and if Zorozo doesn't prove better than any grease you have ever tried, you can return the cups and what is left of the barrel and no charge will be made for the amount consumed.
Or You Might Like This One Better. If So, Write Us
We will send you a trial keg and charge you at the barrel rate, payable in ninety days, and make you a present of 3 "Philadelphia" Compressed Air Grease Cups.
UNIQUE-INCOMPARABLE-ORIGINAL
OXOILOX
"FROM THE OX TO THE OX"
AN EXTREMELY HIGH CLASS BELT DRESSING AND LEATHER PRESERVATIVE.
MANUFACTURED EXCLUSIVELY BY
E.S. WALTON COMPANY
PRESSERS AND REFINERS OF ANIMAL OILS.
PHILADELPHIA, PA. U.S.A.
PHILADELPHIA, PA.
VICTORIA
LIQUID
SOAP
THE
PHILADELPHIA
COMPRESSED AIR
GREASE CUP
PAT. 4-25-05
McLAIN Co. PHILA

ESE LETTERS



Searchlight Mining and Milling Co.

Office: Room 244 I. W. Hellman Building, 4th and Main Sts.
MINES AT SEARCHLIGHT, NEVADA

Searchlight, Nev.,
July 5th, 1911.

Messrs. F. S. Walton Co.,
Orthodox St. & Delaware River,
Philadelphia, Pa.

Dear Sirs:-

I installed a 100 h. p. engine made by the Phoenix Engine Works at Meadville, Pa. about six months ago. The governor is of the ball and spring type, attached to the flywheel. The engine is used to drive a 75 K. W. Generator, which generates power for operating the pumps in the mine and giving lights.

The pin in the governor attachment has given us a great deal of trouble, as it did not get sufficient lubrication and would stick. This caused the engine to race, and made the lights very uneven, besides uneveness in the speed of our pumps. We tried different kinds of grease and oil cups on it, but could not overcome the racing of the engine whenever the pin got dry, which was most of the time.

My head engineer wrote to you for a trial can of your Zorozo Lubricating Grease, about two months ago, and received a grease cup and sample can of said grease, etc. He put the grease cup on the engine, where we had been having the trouble, and since then we have no trouble with the engine racing. Since using your grease the engine has been running as smooth and even as clock work.

I enclose herewith P. O. order for which please send me 100 lb. keg of the Zorozo Grease and an 3 lb. can of OXoilOX Belt Dressing.

Thanking you for the sample sent, and for prompt attention to this order, I am,

Yours truly,
Henry E. Carter
Lessee.

C. B. LAKE & COMPANY, LIMITED
PROPRIETORS CONTRABAND BAYOU CANAL, WESTLAKE RICE MILL
LAKE CHARLES, LOUISIANA
ROBINSON CIPHER

F. S. Walton Co., July 7th, 1911.
Philadelphia, Pa.

Gentlemen:-

Enclosed find draft, covering invoice 6/a/11. Your grease is good grease and no mistake.

Yours truly,

N READ THIS FREE OFFER

NO man can tell how good Zorozo Grease is until he has tried it, so we want you to try it. We have perfect faith in its ability to "make good," and are willing to stand all the expense incident to the trial. If you will do us the favor of filling in and mailing us the attached coupon, we will send you absolutely free, express charges prepaid, a can of Zorozo Grease and a "Philadelphia" Compressed Air Grease Cup to test it in, also a can of OXoilOX Belt Dressing, and a bottle of Victoria Liquid Soap—two more of the products we manufacture. An appropriate quantity of this Victoria Liquid Soap is sent with each order of Zorozo Lubricating Grease and OXoilOX Belt Dressing.

Zorozo and OXoilOX are for sale by many jobbers, but if yours does not carry them send us his name and we'll see that he does.

F. S. WALTON CO.
PHILADELPHIA, PA.

THESE FOUR SAMPLES FREE

COUPON

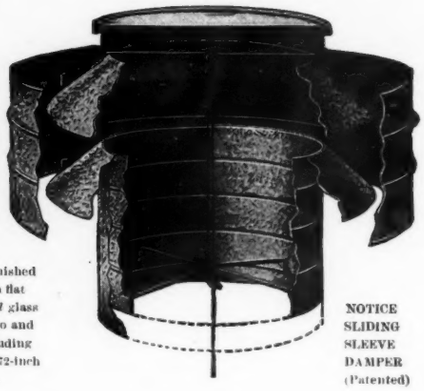
Gentlemen:—You may send me free Zorozo, OXoilOX, Victoria Liquid Soap and Grease Cup.

Name.....
Name of firm.....
Address in full.....
H. P. of engine.....
On what bearing will sample be tested?.....
Its diameter.....
Size of tap for cup.....
R. P. M.....

POWER 9-19-11

U. S. Steel Corporation Places 72nd Order For Burt Ventilators

Metal top with sliding-sleeve damper (patented)



Furnished with flat *wired* glass up to and including the 72-inch size.

NOTICE SLIDING SLEEVE DAMPER (Patented)

Recorders indicate the value of a device—when a concern buys a certain device again and again as occasion arises it is good proof that the device is giving complete satisfaction.

So you can draw your own conclusion from the fact that the U. S. Steel Corporation, the largest company of its kind, has just placed its 72nd order for Burt Ventilators.

Burt Ventilators draw out all impure air, smoke, gas, steam, etc., from the building, without operating expense.

They are superior to other ventilators in that they are made with the adjustable sliding sleeve damper which, having no flat surface, does not collect dust or refuse to be shaken off into the building as with other dampers. The damper is not affected by air currents and requires no attention.

Where glass top Burts are used the damper can be closed, yet light will not be shut off.

Send for our new 112-page Catalog.

The Burt Manufacturing Company

232 Main St., Akron, O., U. S. A.

Largest Manufacturers of Oil Filters in the World.

Geo. W. Reed & Co., Montreal, Sole Manufacturers of "Burt" Ventilators for Canada.